Testing and Fretting Fatigue Analysis of Bridge Stay Cable Wires at Saddle Supports

by

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Author's Declaration

This thesis consists of material all of which I authored or co-authored: see Statement of Contributions included in the thesis. This is a true copy of the thesis, including any required final revisions, as accepted by my examiners.

I understand that my thesis may be made electronically available to the public.

Statement of Contributions

The following contributed to parts of the studies presented in Chapter 5.

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Dr. Liang assisted with polishing approximately half of the hourglass samples used for plain fatigue tests. She also performed most of the plain fatigue tests using hourglass samples. The results of this study are presented in Section 5.8.

Dr. Huda prepared the samples for microstructure analysis, performed the microhardness tests and took the SEM photos shown in Section 5.9.

Writing, commenting on the figures and editing the figures in Sections 5.8 and 5.9 were done by the author; the discussions with Dr. Liang and Dr. Huda and their comments are greatly appreciated.

Abstract

High-strength steel cables are one of the principal components of cable-stayed bridges. In these structures, the cables transfer the gravitational loads at the deck to the bridge tower structures or "pylons". Reliable anchoring of the cables is a primary design consideration for cable-stayed bridges. Traditionally, the cables were directly anchored to the bridge deck and the pylons. A newer approach, employing so-called saddle systems, has become more popular in recent decades. With this approach, the cables are anchored to the bridge deck on one side of the pylon, go over a radial surface at the pylon, and are finally anchored to the bridge deck on the other side of the pylon. Material and anchoring costs of saddle systems are lower than the traditional approach, as smaller pylons are required for saddle systems and the cables do not require anchoring at the pylons. A primary design consideration of saddle systems is fretting fatigue failure of the cables at the saddle supports. Despite this fact, very limited previous research can be found in the literature on this topic. Also, the existing standards for saddle systems are rather simplistic. These standards require large-scale fatigue tests to evaluate the saddle systems and do not offer a calculation-based design procedure. With this in mind, the main objectives of the current thesis are: to undertake initial efforts to develop a calculation-based framework to evaluate the fretting fatigue behaviour of cables at saddle supports, to explore a possible framework for probabilistic analysis of this problem, and to design a more economical small-scale fretting fatigue test setup and use it to evaluate the fretting fatigue behaviour of typical bridge cable wires.

Several parameters affect the fretting fatigue behaviour of cables (e.g., the relative displacement between the cable and saddle, and the contact force between the cable and saddle). In this thesis, closed-form equations for evaluating these critical parameters are first discussed. Then, an FE model is developed to evaluate the accuracy of these equations. The developed FE model is then used to evaluate the effect of wear on these critical parameters. Overall, the results of the FE model are shown to be close to the results obtained by calculation. However, a higher difference is seen between the results at the points where the cable first meets the saddle. Following the determination of the critical parameters, a multiaxial stress approach based on the Smith-Watson-Topper (SWT) parameter is used to evaluate the fretting fatigue life of the cable wires. A set of large-scale tests previously performed at TU Berlin is used as an example. The predictions based on the SWT

parameter are shown to be in good agreement (i.e., fatigue lives and overall trends are estimated with reasonable accuracy) with the tests performed at TU Berlin.

In order to extend this approach to a probabilistic framework, several practical approaches aimed at limiting the need to perform time-consuming FE analyses are then explored. These approaches include the use of Monte Carlo simulation (MCS) with fretting maps or employing the multiplicative dimensional reduction method (M-DRM). The results of these approaches are then compared, and the challenges and benefits of each approach are presented. The results obtained using both methods are reasonably close to each other. Finally, an analysis is performed to evaluate the sensitivity of the prediction results to the main model parameters. It is shown that the uncertainties in the contact force and fatigue strength coefficients have the highest sensitivity factors.

Following the completion of these analytical studies, a small-scale fretting fatigue test setup was designed to evaluate the fretting fatigue behaviour of bridge stay cable wires. Two different bridge cable types, namely: galvanized and bare, were used for these tests. In these tests, the critical parameters affecting fretting fatigue life were varied. It was found that the bare wires have a better fretting fatigue performance in comparison with the galvanized wires. Following the completion of the fretting fatigue tests, plain fatigue tests were performed to evaluate the fatigue performance of the wire material. After the experimental work, a microstructure analysis was performed to evaluate the microhardness of the wires and observe defects at the surface and core of the wires using SEM photography. Irregular microstructures were found at the surface of the galvanized wire. However, the bare wire had a uniform microstructure at the surface.

Following the experimental work, the SWT parameter-based approaches were applied to the tests performed at the University of Waterloo. However, these approaches have limitations in that they do not account for wire defects and their influence on the fatigue life predictions. Given the presence of significant defects in the wires, a linear elastic fracture mechanic (LEFM) approach is lastly employed to study possible effects of these defects on the fretting fatigue life of the wires. The LEFM results are shown to be in good agreement with the test results.

Acknowledgements

I would like to express my deepest appreciation to my supervisor, Scott Walbridge, who has always been supportive and encouraging. I could not have undertaken this journey without his insightful comments and ideas.

I would like to sincerely thank Dr. Al-Mayah, Dr. Jahed, Dr. Pandey, and Dr. Wegner for agreeing to be on my committee, reviewing my work, and providing valuable comments.

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I wish to greatly thank my parents, Borzoo and Shirin, my spouse, Firoozeh, and my sister, Marjan, who have always supported me through my ups and downs.

I would like to thank my research group members, Laurent Gerin, Melanie Perreault, Carol Liang, Rakesh Ranjan, Matthew Sjaarda, Michelle Chien, Spencer Arbuckle, Mohammad Barkhori, Abdullah Abdelbadie, Mi Zhou, and Mohammad Tolou Kian who made my experience more delightful at UW. They were always available for discussions related to research and were willing to comment on my works and presentations. I greatly appreciate their support.

Last but not least, I would like to thank my friends in Waterloo, whom I spent most of my leisure time with, Amirmahdi Mostofinejad, Ali Ghavidel, Daniel Rafii, Hossein Soleimani, Hamid Daryan, Amir Safarpour, and Momo Zandieh.

Dedication

I would like to dedicate this thesis to Mansour Esnaashary Esfahani. He was a great friend and a hard-working student. He was onboard Flight 752. He will always be remembered.

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1. Introduction

1.1 Fretting fatigue

Fretting fatigue occurs "when two contacting components experience a small amplitude relative motion" (Ding et al. 2011). Figure 1-1 shows a schematic view of a fretting fatigue problem. As shown in Figure 1-2, three different regimes can be defined for two contacting components (Vingsbo and Soderberg 1988): a stick regime at very small relative displacements (0-2 μm) with a low volume of wear and a relatively high fatigue life, a stick-slip regime at higher relative displacements (2-20 μm) with a stick regime at the center of the contact area and a slip regime at the border of the contact area corresponding with a lower fatigue life, and finally a gross sliding regime at higher relative displacements with a high degree of wear over the entire contact surface typically resulting in a higher fatigue life than that associated with the stick-slip regime. Fatigue life is greater in the gross sliding regime because a high degree of wear can remove small cracks propagating at the surface of the components.

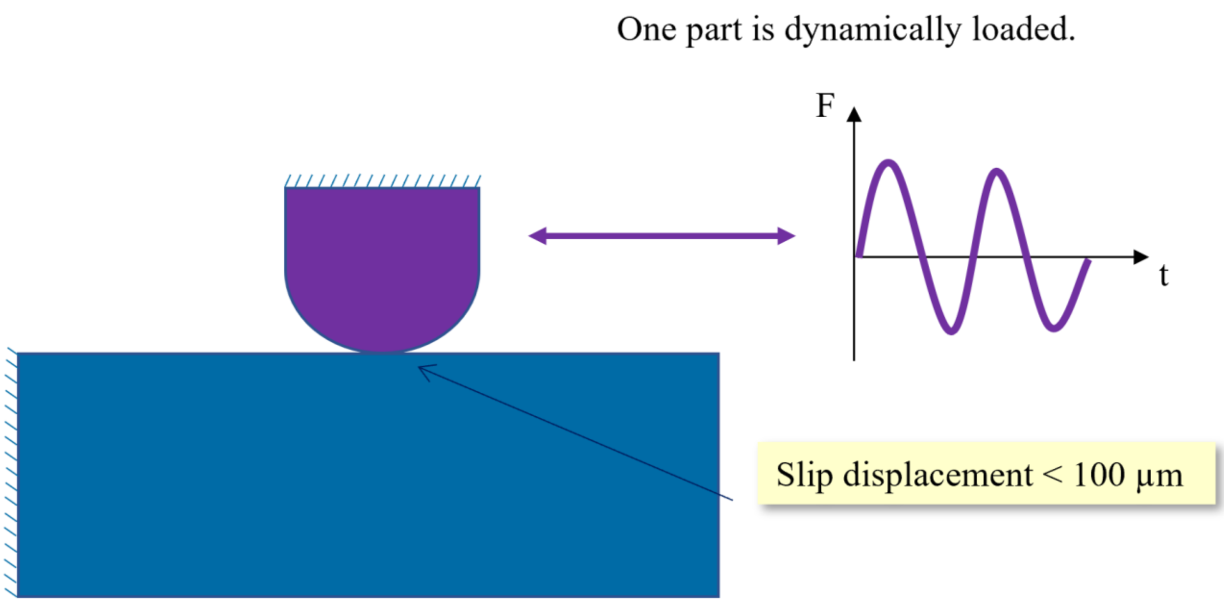


Figure 1-1 Schematic view of a fretting fatigue problem.

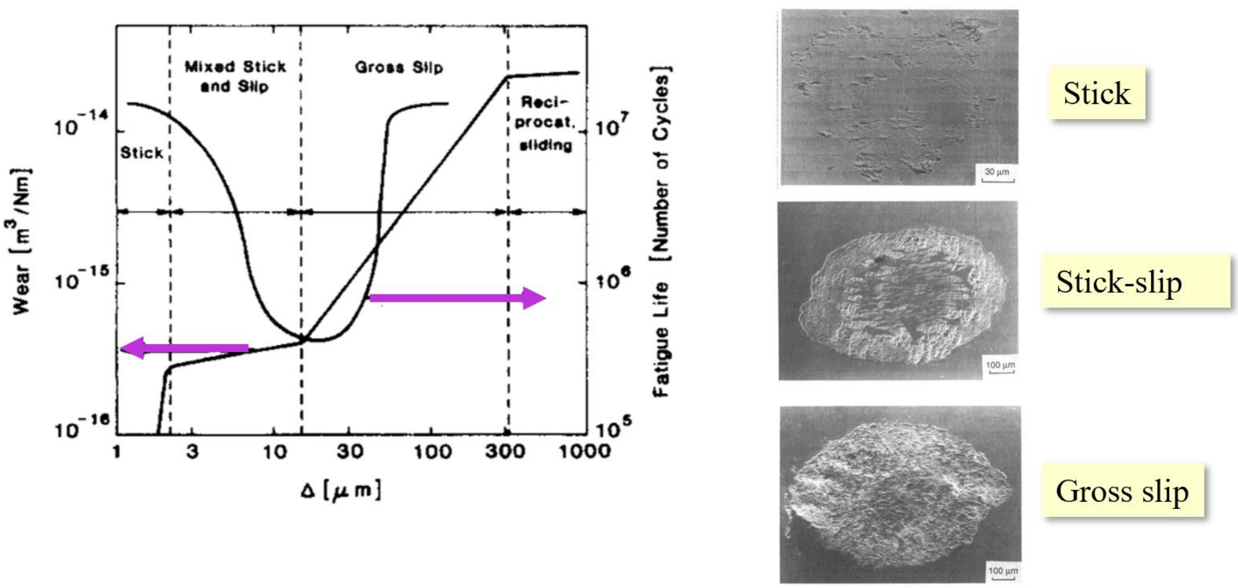


Figure 1-2 Fretting maps by (Vingsbo and Soderberg 1988).

Based on the definition of fretting fatigue in the previous paragraph, fretting fatigue failure can be a serious issue for connections and machines where two parts are in contact, and one part is cyclically loaded. Examples of applications that can be prone to fretting fatigue are bolted and riveted connections, dovetail connections, bridge cables, and bearing shafts.

1.2 Fretting fatigue problem in cable-stayed bridges

In cable-stayed bridges, high-strength steel cables transfer the gravitational loads at the bridge deck to the bridge tower structures, called pylons. Reliable anchoring of the cables is a primary design consideration for cable-stayed bridges. Traditionally, the cables were directly anchored to the bridge deck and the pylons. A newer approach with lower material and anchoring cost, employing so-called saddle systems, has become more popular in recent decades. With these systems, the cables are anchored to the bridge deck on one side of the pylon, go over a radial surface (saddle) at the pylon, and are finally anchored to the bridge deck on the other side. Figure 1-3 (a) shows a cable-stayed bridge employing saddles to support the cables at the pylons. A schematic view of a cable over a saddle is shown in Figure 1-3 (b).

As already discussed, fretting fatigue occurs when there is a small relative displacement between two contacting components. As the trucks cross the bridge, the cables can slightly move along the saddle. The high contact forces at the contact points between the cable wires and the saddle, in addition to the small relative displacements between the cables and the saddle, make the cables prone to fretting fatigue failure, which is, therefore, an important design consideration.

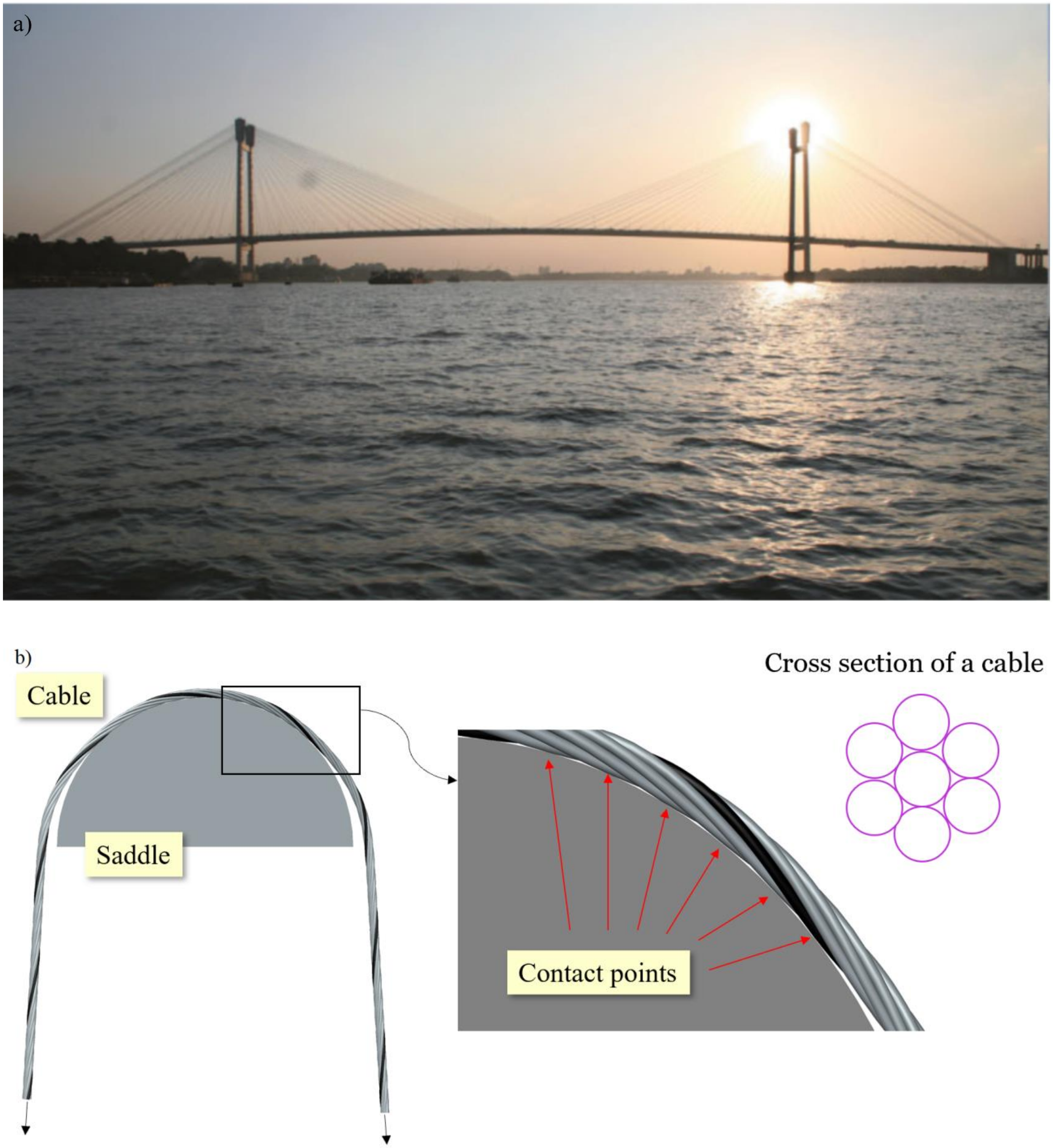


Figure 1-3 Example of a cable-stayed bridge (a) from Schlaich et al. (2012), schematic view of a cable over a saddle (b).

1.3 Motivation

While fretting fatigue failure is a major design consideration for saddle systems, there is limited work on this topic in the literature. The current design provisions (fib 2005, PTI 2012) do not include calculation-based procedures and require companies to do large-scale tests of saddle systems to evaluate the fretting fatigue performance of cables in these systems. The current provisions are very simplistic. Specifically, the following issues were found:

- Very little can be found in these standards regarding the reliability of the saddle systems. These standards require a limited number of tests to evaluate these systems and do not require tests to failure. Therefore, they do not offer a reliability level consistent with the conventional design codes for other structural elements on a bridge.
- The current standards specify the same proof testing procedure for all anchoring systems. They do not include specific details needed for the design of different systems, and they do not discuss the parameters that affect the fretting fatigue life of saddle systems.
- The current provisions are based on constant amplitude loading and do not include any information on variable amplitude loading or discuss its effect on fretting fatigue performance. In real bridges, the cables are under variable amplitude loading conditions.

1.4 Objectives

Against this background, the main objectives of this project are as follows:

- *To determine the critical parameters that affect the fretting fatigue life of cables (e.g., contact force, slip displacement):* There are several critical parameters that affect the fretting fatigue behaviour of cables at saddle supports. There are a few closed-form equations and methods presented in previous works to calculate suitable values for these parameters. These equations were developed with simplifying assumptions and have not been proven to be accurate for the problem of a cable over a saddle. Therefore, the first objective of the project is to review the methods available to calculate the critical parameters and compare their outputs with the results obtained by finite element (FE) analysis.

- *To predict the fretting fatigue life of steel stay cables using a calculation-based method*: As discussed before, there is a limited number of works on this topic in the literature and a calculation-based design procedure cannot be found in the provisions and standards for saddle systems. The second objective of the current study is therefore to predict the fretting fatigue life of the cables using a numerical analysis or a calculation-based method.

- *To determine the effect of wear on critical parameters between the cable and saddle:* Given that the cable is harder than the saddle, a high amount of wear, typically, can be seen on the saddle part. Due to this wear, the contact forces might change at the contact points between the cables and the saddle. Determining possible effects of wear on critical parameters and then the fretting fatigue life is therefore another objective of the current study.
- *To develop and apply probabilistic frameworks for fretting fatigue analysis of bridge stay cables at saddle supports*: No work can be found on probabilistic analysis of this problem. Therefore, evaluating different possible frameworks for probabilistic analysis of this problem is another objective of this project. Based on estimations of the uncertainties associated with the various model parameters, a probabilistic analysis should be performed to obtain the fatigue life statistical distribution. Such distributions would allow risk-based design criteria to be established, thus enabling the design of saddle supports using an approach philosophically consistent with the current structural design standards.
- *To design a small-scale fretting fatigue test setup*: A few works can be found in the literature on large- or full-scale tests of saddle systems. However, no small-scale setup can be found in the literature to evaluate the fretting fatigue behaviour of cables at saddle supports. Small-scale tests are more economical, typically quicker, and more efficient. Therefore, designing a small-scale test for this problem is another objective of this thesis.
- *To determine the effects of varying the critical parameters on the fretting fatigue behaviour of cable wires*: Once the small-scale test is designed, the next step will be to assess the fretting fatigue performance of bridge cable wires using this small-scale fretting fatigue setup. In this way, the effects of varying critical parameters on the fretting fatigue behaviour of cable wires will be studied experimentally.
- *To evaluate the material properties of the bridge cables and establish the input parameters for the deterministic and probabilistic analysis frameworks:* Evaluating the material properties of the cables including microhardness, tensile and fatigue properties (Coffin-Manson parameters)

is the last objective of this thesis. A goal of this work will be to establish the input parameters for the developed calculation-based and probabilistic analysis methods.

1.5 Overview of the thesis

This thesis starts with a literature review, which discusses previous studies on fretting fatigue problems. Different methods for predicting the fretting fatigue life of structural components are discussed. Following that, the previous works on bridge cables are discussed. In the end, some of the previous small-scale fretting fatigue tests for other applications are presented.

Chapter 3 starts by comparing the results of the methods developed previously in the literature for evaluating the critical parameters at the contact points along the length of a bridge cable with results obtained using a 2D FE model developed in the current thesis. Following this, a 3D FE model of a single contact point is used to make fretting fatigue life predictions. A multiaxial stress approach with the Smith-Watson-Topper (SWT) parameter is used for this analysis. Previous works on wear modelling have focused on wear effects at single contact points in mechanical connections. The current problem is somewhat unique in that it involves multiple contact points, and one effect of the wear is that it can result in a redistribution of the forces at each contact point. With this in mind, in the last part of this chapter, the effect of wear on the critical parameters at the contact points and the fretting fatigue life of the cables is evaluated using the 2D FE model.

Chapter 4 first reviews Monte Carlo Simulation (MCS) and the Multiplicative dimensional reduction method (M-DRM) for probabilistic analysis of structural performance. A probabilistic framework for assessing fretting fatigue survival probability based on MCS is then discussed. However, given the high computational time due to the nonlinear 3D FE analysis required for each MCS trial, this framework is not practical. Therefore, possible alternatives using MCS along with fretting maps and M-DRM are subsequently investigated in the current study. Lastly, a sensitivity analysis is performed to determine the critical variables of the problem.

Chapter 5 first presents the small-scale fretting fatigue setup designed for the current project at the University of Waterloo, then goes over the fretting fatigue test results in detail and compares the results for two different wire types: bare and galvanized. Following this, the properties of both wire types including microhardness, tensile, and fatigue (specifically Coffin-Manson parameters) are determined using hourglass (machined) wire samples. Lastly, wire defects at the surface and the core are detected using scanning electron microscope (SEM) photography.

Chapter 6 applies the methods developed in Chapters 3 and 4 to the tests performed at the University of Waterloo. It first evaluates the results based on a deterministic analysis and discusses possible challenges of using the SWT parameter for comparing the fretting fatigue life of wires given the fact that defects were found in both wires. Following that, probabilistic frameworks based on the MCS and MDRM were used to determine survival probabilities for tests performed at the University of Waterloo.

Based on the fretting fatigue test observations and SEM photos in Chapter 5, it is concluded the defects at the surface/core of the wires can significantly affect the fatigue life of the wires. In Chapter 7, linear elastic fracture mechanics is therefore used to predict the fretting fatigue life of the wires, allowing the defect size to be considered as one of the analysis inputs. In this analysis, weight functions are used to determine stress intensity factor ranges. Following this, the Paris-Erdogan law is used to determine the fretting fatigue life of the wires.

Chapter 8 presents the main results and contributions of the research presented in the current thesis. It then discusses possible areas of future work on this topic.

2. Literature Review

2.1 Fretting fatigue

Fretting fatigue occurs "when two contacting components experience a small amplitude relative motion" (Ding et al. 2011). The primary reason fretting fatigue occurs, is because the contact causes a stress concentration and the cyclic loading causes the stress concentration to move, which means if one tracks the stress at a single point near the stress concentration, they will see that it fluctuates. A fluctuating stress is necessary for all varieties of metal fatigue according to the basic definition of "fatigue". A schematic view of a typical fretting fatigue test setup is shown in Figure 2-1. Hills et al. (1988) and Nowell et al. (2006) presented the basic principles of the fretting fatigue problem and discussed the main parameters that affect the fretting fatigue life of components. The elements at the surface of the contacting components are critical locations for crack initiation and propagation due to the stress concentration in these elements. Different fretting fatigue setups have been developed to study fretting fatigue problems. These setups typically consist of one or two contacting pads and a flat specimen. The contacting pads apply the contact force to the specimen, and the specimen is cyclically loaded between maximum and minimum stress levels. Fretting fatigue tests at different stress ranges, contact forces, and slip displacements show that the specimens typically fail at the edge of the contact area.

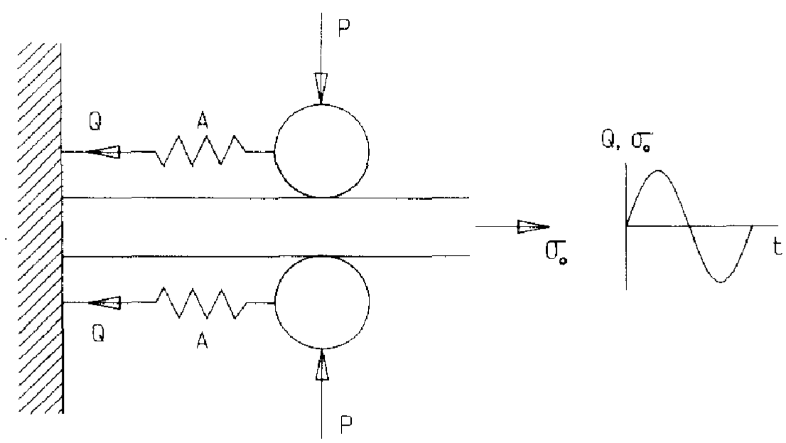


Figure 2-1 Schematic view of fretting fatigue tests (Hills et al. 1988).

Vingsbo and Soderberg (1988) first presented the concept of "fretting maps" for fretting fatigue problems. In these maps, the effects of varying different parameters on the fretting fatigue life of a component are evaluated. In this study, three different regimes were identified for a fretting fatigue problem (see Figure 2-2): a stick regime at very small relative displacements (0-2 μm) with a low volume of wear and a relatively high fatigue life; a stick-slip regime at higher relative displacements (2-20 µm) with a stick regime at the center of the contact area and a slip regime at the border of the contact area corresponding with a lower fatigue life; and finally a gross sliding regime at higher relative displacements. There is a high degree of wear over the entire contact area in the gross sliding regime. The fatigue life typically increases in this regime with increasing the slip displacement. This can be explained by the fact that a high degree of wear can actually remove small cracks and stop them from propagating, thus increasing fatigue life.

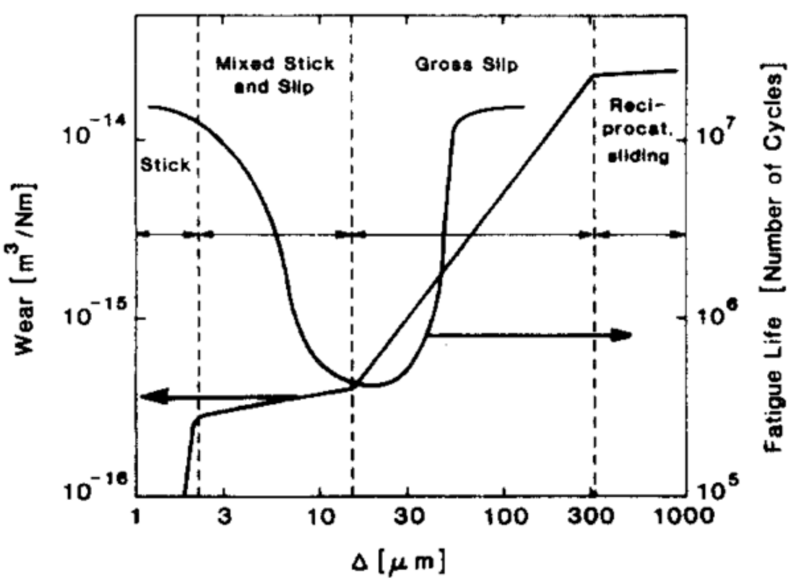


Figure 2-2 Fretting map example from Vingsbo and Soderberg (1988).

Nowell et al. (2006) reviewed the previous studies on this topic up to 2006. They discuss fretting fatigue prediction methods based on multiaxial stress analysis. This study also discusses the reasons for higher fatigue life at higher slip displacements (typically greater than 50 µm). Two possible explanations are presented for this trend. First, the wear debris between the two contacting components can act as a lubricant and decrease the coefficient of friction and consequently the stress range for the elements at the surface. The second explanation, which has received more attention, is a high degree of wear can remove the small cracks propagating at the surface of the specimens. Furthermore, it is discussed that the multiaxial stress approach should be used over a volume (i.e., the average of the SWT parameter should be determined using the SWT parameter values at several points inside the volume). This volume should be determined so as to achieve a good prediction of the test results. Following this, another approach based on relating the stress field of the fretting fatigue problem to the stress field of a notched plate are discussed. Lastly, the short crack arrest problem is discussed and the Kitagawa-Takahashi diagram is used for crack arrest analysis.

2.2 Critical parameters affecting the fretting fatigue life of components

There are a number of parameters that can affect the fretting fatigue life of components. It has been reported that up to fifty variables can affect the fretting fatigue behaviour of components (Dobromirski 1992). The effect of some of them, however, is not significant. The most important ones are slip displacement, stress range, contact force, and coefficient of friction. In this section, the effects of these parameters on fretting fatigue life are discussed.

2.2.1 Slip displacement

As discussed earlier, Vingsbo and Soderberg (1988) presented three different regimes based on the slip displacement: stick, stick-slip (or partial slip), and gross sliding. Based on this work and others (e.g., Gao et al. 1991, Jin and Mall 2004), the fretting fatigue life is typically high in the stick regime at very small relative displacement. Then the fretting fatigue life decreases by increasing the relative displacement at the stick-slip regime. Finally, the fretting fatigue life increases in the gross sliding regime as the high amount of wear can remove the micro-cracks propagating at the surface of the components. This trend can be seen in Figure 2-2 and Figure 2-3.

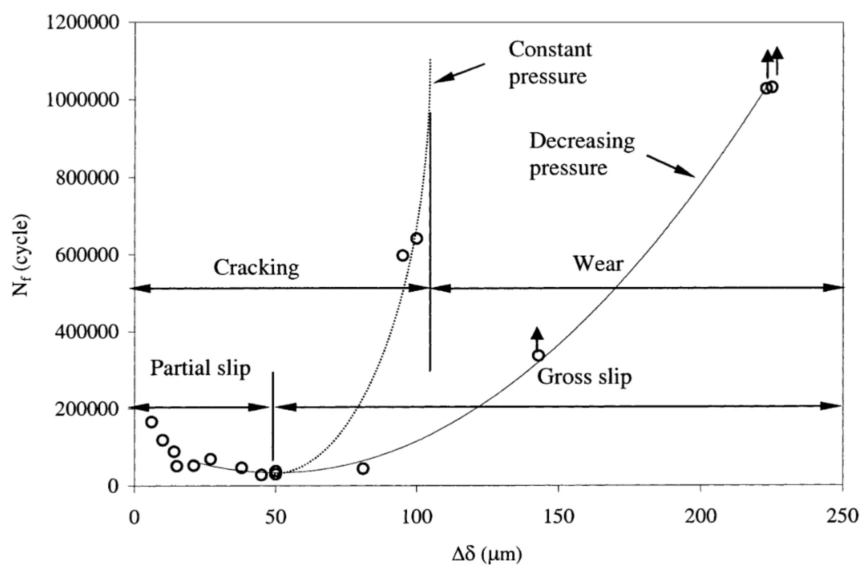


Figure 2-3 The effect of slip displacement on fretting fatigue life (Jin and Mall 2004).

2.2.2 Contact force

Contact force or contact pressure is another parameter that affects the fretting fatigue life of contacting components. Looking at Figure 2-4, three different regimes can be seen by varying the contact pressure or contact force (Vingsbo and Söderberg 1988). At low contact forces, a gross sliding regime with a high volume of wear can be seen as the components can easily move along each other. With increasing the contact force, a stick-slip regime can be seen with a stick regime at the center and a slip regime at the border. With further increasing the contact force, a stick regime can be observed with a limited amount of wear. Similar to the trend discussed previously for the effect of slip displacement on fretting fatigue life, the minimum fretting fatigue life occurs in the stick-slip regime and higher fatigue life can be seen in stick and gross sliding regimes.

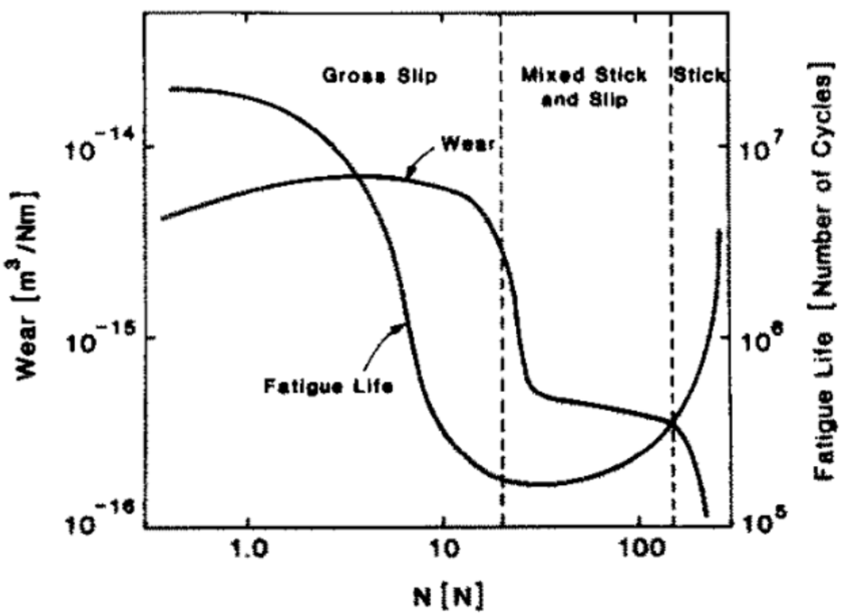


Figure 2-4 The effect of contact force on fretting fatigue life (Vingsbo and Söderberg 1988).

2.2.3 Stress range

Stress range is a primary parameter that affects the fatigue life of components. In general, the fatigue life of components decreases with an increase in the stress range. A similar trend has been seen for fretting fatigue tests as well (Iyer and Mall 2001, Perier et al. 2009, Wang et al. 2011). Higher stress range is associated with a shorter fretting fatigue life. The reason is also the same as for fatigue under direct loading – if the global or "nominal" (far-field) stress range is higher, then so will be the cyclic stress concentration causing the local fatigue damage.

2.2.4 Coefficient of friction

The coefficient of friction, COF, is another primary parameter affecting the fretting fatigue life of components. This parameter typically increases during fretting fatigue tests as the COF of the worn surfaces is higher than the COF of the initial smooth surfaces. Figure 2-5 shows the increase in the COF during the first few cycles of fretting fatigue tests presented in Hills et al. (1988).

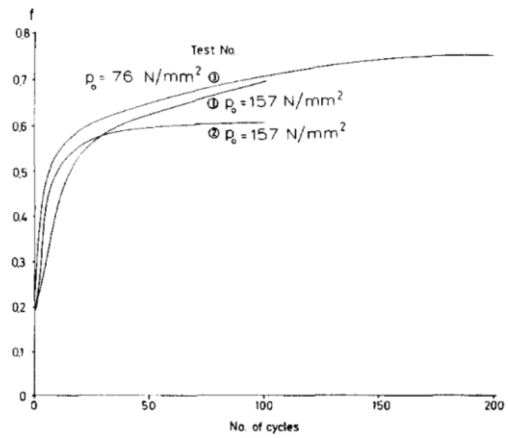


Figure 2-5 The increase in COF during fretting fatigue tests (Hills et al. 1988).

The measurement of an accurate COF in fretting fatigue tests is very challenging. Given the different amounts of wear at the different points at the contact surface (especially, in the stick-slip regime), only an average COF can be practically measured during the tests, and even measuring the average COF can be very challenging for tests in the stick-slip regime.

Figure 2-6 shows a block over a flat surface, and the forces acting on this block. In the gross sliding condition, the applied force is higher than the frictional force (or COF times the contact force ($\mu \cdot N$)). Therefore, there is slip displacement all over the contact surface and an average COF can be calculated using the measured frictional force as follows:

$$COF_{ave} = \frac{Frictionalforce\ (F_f)}{Contact\ force\ (N)} \qquad 2\text{-}1$$

In stick and stick-slip regimes, however, measuring the COF during the tests is very challenging. In these regimes, the frictional force is lower than the maximum capacity of the frictional force at the surface ($\mu \cdot N$). Therefore, dividing the frictional force by the contact force is only a normalized frictional force or quasi-COF as described in Jin and Mall (2004). It should be noted that an increase in COF has also been seen in tests in stick-slip (not only in gross slip), as there is wear at the border of the contact area in this regime, which also impacts COF.

$$quasi - COF = \frac{Frictionalforce\ (F_f)}{Contact\ Force\ (N)} \qquad 2\text{-}2$$

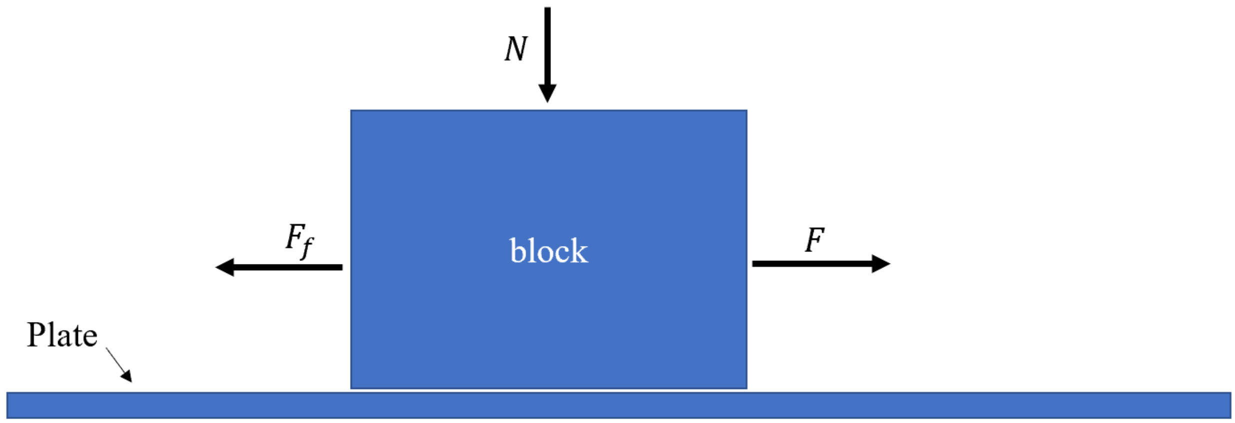


Figure 2-6 Schematic view of a block over a plate.

Murthy et al. (2006) presented a test procedure for measuring the COF in the stick-slip regime. This COF test can be done after a few thousand cycles of a fretting fatigue test. In this test, the stress range keeps increasing until the specimen moves along the contacting component. They propose that this point is associated with a sharp sound, and the force required for moving from the stick-slip regime to the gross sliding regime can thus be recorded and used for COF calculation. One issue with this method is it essentially consumes a specimen, which cannot be used subsequently to obtain a fatigue test result, as once gross slip occurs, it will coincide with wear, which will influence the result of the original stick-slip test. Of course, a friction test at the end of the fatigue test is not possible, as at this point the loaded component has fractured.

2.3 Fretting fatigue analysis using multiaxial stress-based approaches

Araújo and Nowell (2002) present a well-known study on the multiaxial stress-based analysis of fretting fatigue problems. In this paper, the Smith-Watson-Topper (SWT) and Fatemi-Socie (FS) parameters are employed. Titanium and aluminum samples are analyzed. In this study, the average SWT and FS parameter values over a critical length, d_c, are used for determining the fretting fatigue life (see Figure 2-7). Trends in the results based on the two studied parameters (SWT and

FS) were similar. The main reason for using an average SWT over a length or volume is to account for the high stress gradient at the surface and possible short crack arrest in fretting fatigue problems. One of the challenges in fretting fatigue life prediction based on this averaging method is determining the length that is required for the calculation. In this study, several lengths were used for calculating the average of the parameters; the length that best fits the test results can be assumed to be the appropriate length for further analysis (see Figure 2-8).

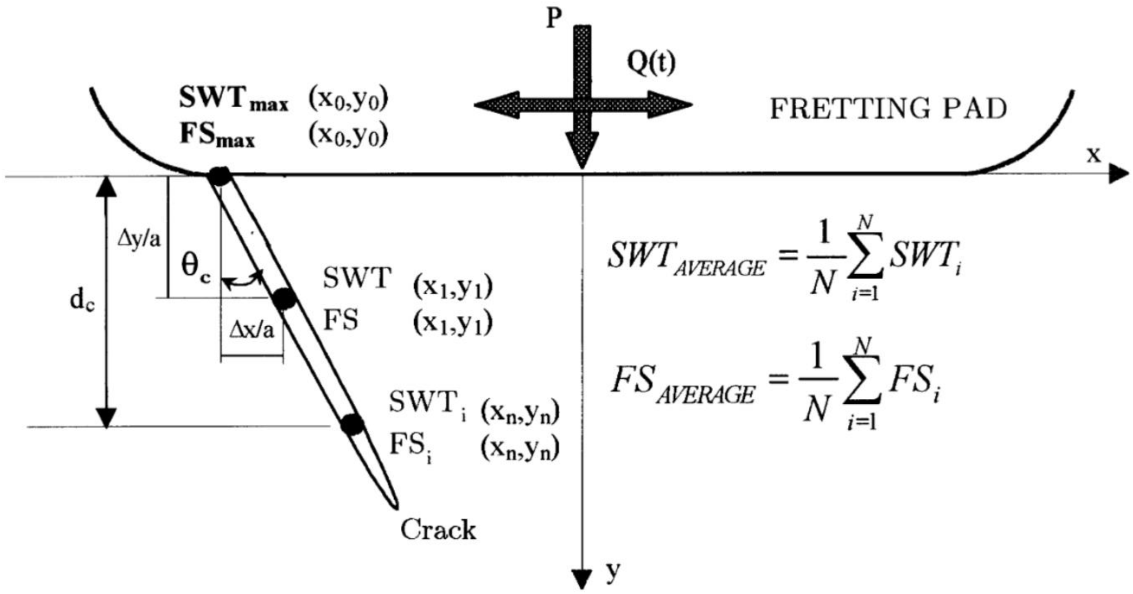


Figure 2-7 Averaging method for SWT and FS parameters (Araújo and Nowell 2002).

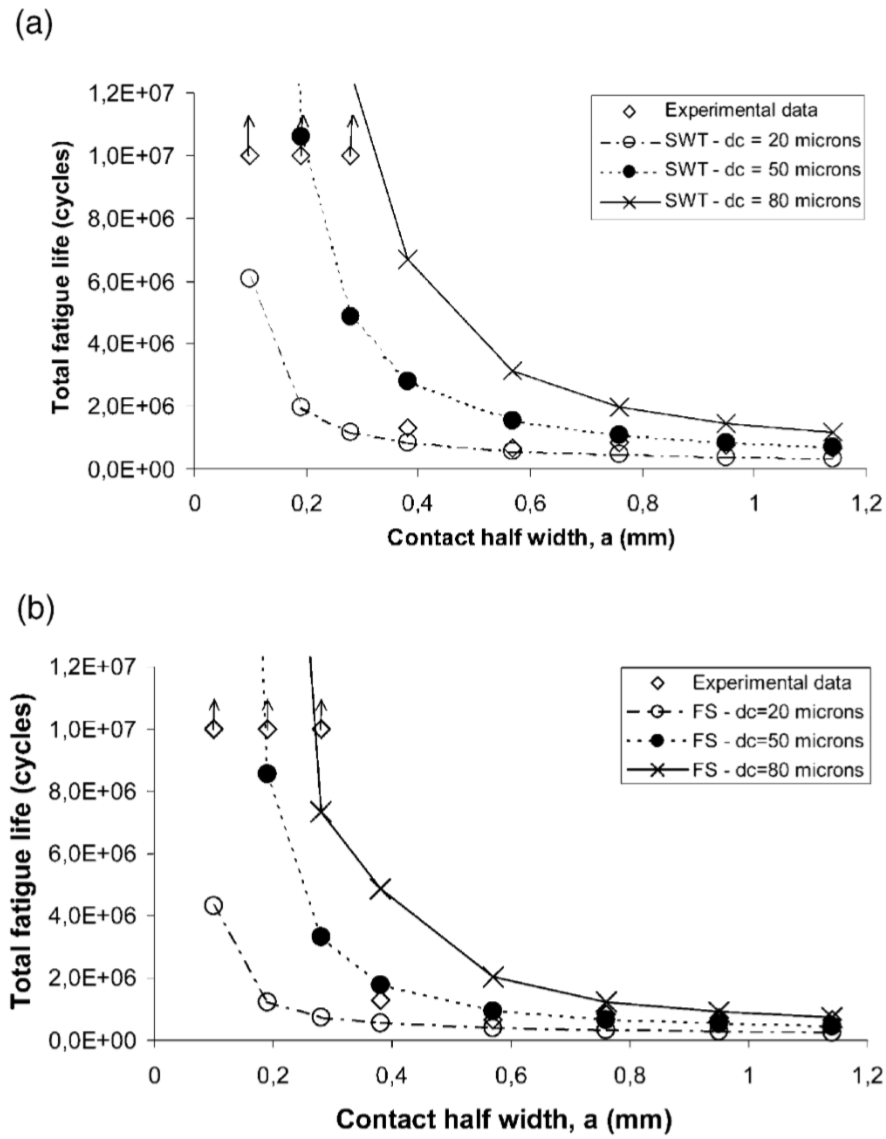


Figure 2-8 Sample results based on SWT and FS parameters versus the test results with different lengths, d_c, assumed for calculating the average parameters (Araújo and Nowell 2002).

Bernardo et al. (2006) proposed an alternative approach to consider the size effect and the effect of high stress gradient in this problem. Based on their approach, the element size (i.e., process zone) can be directly used instead of averaging volume. This method is computationally efficient as coarser elements can be used in the contact area. Similar to the averaging method, the challenge in using this approach is determining the appropriate element size. The proposed method is determining the element size in a way that the test results can be predicted reasonably well based on the output of the FE analysis.

The theory of critical distance also was used in several papers in the field to account for the stress gradient. This theory was first developed for notches (e.g., Taylor 2004, Zhu et al. 2020); however, given the similarity of the stress gradients at the edge of the contact area of fretting fatigue problems with notches, this method was used as another approach for considering the effect of stress gradients in fretting fatigue problems. This method can be applied using the line or point method. In the line method, the average of the damage parameter should be calculated over a critical length, L. In the point method, the damage parameter should be calculated at a point with a depth of L/2 from the hot spot (edge of the contact in fretting fatigue problems). L can be calculated using the following equation:

$$L = \frac{1}{\pi}\left(\frac{\Delta k_{th}}{\Delta \sigma_0}\right)^2 \qquad 2\text{-}3$$

where Δk_{th} is the threshold stress intensity factor and $\Delta \sigma_0$ is the fatigue limit of the component in fully reversed fatigue tests. This method seems to be a good approach to capture the effect of high stress gradients.

Recently, a new approach has been developed for notch problems to handle mesh refinement issues in computationally expensive 3D models, called the theory of critical distance with mesh control (Vargiu et al. 2017). In this approach, first, a coarse mesh will be used for the analysis of the problem. Following that, instead of measuring the damage parameter at a depth of L/2, the damage parameter at the hot spot will be calculated and used for the analysis. Based on the analysis in this paper, an element size of 2.87L should be used for the coarse FE model. Following this work, Zabala et al. (2020) and Infante-Garcia et al. (2022) applied this method to the fretting fatigue problems and determined the optimal element size to be between 1.6L to 2.6L. The relative errors are reported for different element sizes. All in all, this method seems to be similar to the one previously discussed by Bernardo et al. (2006); however, it uses a clearer definition for the element size.

Sum et al. (2005) discussed how the SWT parameter can be used for determining the fretting fatigue life of the components in 2D and 3D problems. This study summarised the required stress/strain transformation required for determining stress and strain ranges in any plane of a 2D or 3D element. Based on this work, stresses and strains in 2D for a given angle, θ_i, can be calculated as follows:

$$\sigma_{11'} = \frac{\sigma_{11}+\sigma_{22}}{2} + \frac{\sigma_{11}-\sigma_{22}}{2}\cdot\cos(2\theta_i) + \tau_{12}\cdot\sin(2\theta_i) \qquad \text{2-4}$$

$$\varepsilon_{11'} = \frac{\varepsilon_{11}+\varepsilon_{22}}{2} + \frac{\varepsilon_{11}-\varepsilon_{22}}{2}\cdot\cos(2\theta_i) + \varepsilon_{12}\cdot\sin(2\theta_i) \qquad \text{2-5}$$

In 3D, the equations are as follows:

$$\sigma = \sigma_{11}\cdot n_x^2 + \sigma_{22}\cdot n_y^2 + \sigma_{33}\cdot n_z^2 + 2\cdot\tau_{12}\cdot n_x\cdot n_y + 2\cdot\tau_{23}\cdot n_y\cdot n_z + 2\cdot\tau_{13}\cdot n_x\cdot n_z \qquad \text{2-6}$$

$$\varepsilon = \varepsilon_{11}\cdot n_x^2 + \varepsilon_{22}\cdot n_y^2 + \varepsilon_{33}\cdot n_z^2 + \gamma_{12}\cdot n_x\cdot n_y + \gamma_{23}\cdot n_y\cdot n_z + \gamma_{13}\cdot n_x\cdot n_z \qquad \text{2-7}$$

where: $n_x = -\sin(\theta_v)\cdot\sin(\theta_h)$, $n_y = \cos(\theta_h)$, $n_z = -\sin(\theta_h)\cdot\cos(\theta_v)$

where θ_h and θ_v are shown in Figure 2-9. These angles can be varied in 5° increments to determine the critical plane with the maximum value of the SWT parameter.

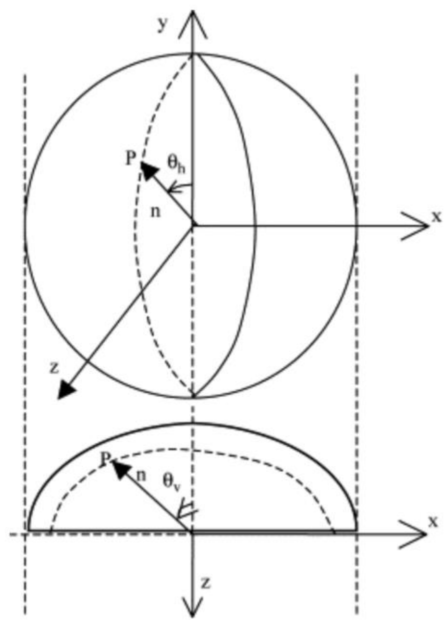


Figure 2-9 θ_h and θ_v for 3D stress/strain transformations (Sum et al. 2005).

Wear can remove small cracks and increase the fretting fatigue life. Also, it can reduce the stress concentration and thus be shown to be beneficial in fretting fatigue analysis evaluation using a multiaxial stress approach. With modelling the wear, the contact area between the components increases, and consequently, the contact pressure decreases. Lower contact pressure means lower stress concentration at the critical points in the contact area. Modelling wear can be done using an iterative approach in which a first FE analysis is done and then based on the results, a portion of the elements at the contact surface can be removed and a new model generated. This process can repeat several times based on a step size (number of cycles for each iteration). Madge et al. (2007a) and Madge et al. (2007b) employed the Archard equation to model wear in a FE analysis. The test results of Jin and Mall (2004) were used in this work to evaluate the accuracy of the developed method. The wear was modelled using a numerical version of the Archard equation as follows:

$$\Delta h(x,\tau) = \Delta N \cdot k_1 \cdot p(x,\tau) \cdot \delta(x,\tau) \tag{2-8}$$

in this equation, Δh is the wear depth, ΔN is the number of cycles in each increment, k_1 is the wear coefficient, p is the contact pressure at the point of interest, and δ is the slip displacement at the point of interest. Based on this work, modelling the wear increases the fatigue life, especially in the gross sliding regime.

Modelling the wear using the Archard equation and the frameworks discussed in Madge et al. (2007a) and Madge et al. (2007b) is complex and time-consuming. Based on the number of cycles in each step, multiple FE analyses are required for only one fatigue life prediction. Ding et al. (2011) presented a simpler approach to consider the beneficial effect of wear. In this work, a correction parameter (called D_{fret2}) was multiplied by the SWT parameter to consider the effect of wear. This parameter is defined as follows:

$$D_{fret2} = (1 + C \cdot \tau \cdot \delta) \cdot \left\langle 1 - \frac{\tau \cdot \delta}{(\tau \cdot \delta)_{th}} \right\rangle^n \tag{2-9}$$

where $\tau \cdot \delta$ is the frictional work during one cycle, C and $(\tau \cdot \delta)_{th}$ are material properties, which can be determined by fretting fatigue tests and then fitting the model predictions to the test results. In this work, the SWT parameter is multiplied by D_{fret2}, and the number of cycles to failure, N_f, is calculated as:

$$\sigma_{\max} \cdot \Delta\varepsilon_a \cdot D_{fret2} = \frac{(\sigma'_f)^2}{E} \cdot (2 \cdot N_f)^{2 \cdot b} + \sigma'_f \cdot \varepsilon'_f \cdot (2 \cdot N_f)^{b+c} \text{ for } \tau \cdot \delta \le (\tau \cdot \delta)_{th} \tag{2-10}$$

The unmodified SWT should be used when $\tau \cdot \delta > (\tau \cdot \delta)_{th}$. In this study, an averaging method, similar to the ones explained in Araújo and Nowell (2002) was employed.

2.4 Fretting fatigue analysis using fracture mechanics and mixed approaches

A short crack fretting fatigue analysis was presented in Araújo and Nowell (1999). According to this study, cracks initiate quickly at the surface due to high stress gradients in fretting fatigue problems. However, they may arrest if the far-field stresses are small. A Kitagawa-Takahashi diagram as shown in Figure 2-10 was employed in this study. Based on this work, the threshold stress range below which crack propagation does not occur starts from zero for very small cracks and increases to a constant value. It should be noted, however, that the stress range should be more than the fatigue limit for the cracks to grow (see Figure 2-10 (b)). Based on this work, the threshold stress intensity factor range for long cracks can be calculated as:

$$\Delta K_0 = 1.12 \cdot \sigma_{fl} \cdot \sqrt{\pi \cdot b_0} \qquad \text{2-11}$$

where σ_{fl} is the constant amplitude fatigue threshold and b_0 is the critical crack length that defines the boundary between a short and a long crack.

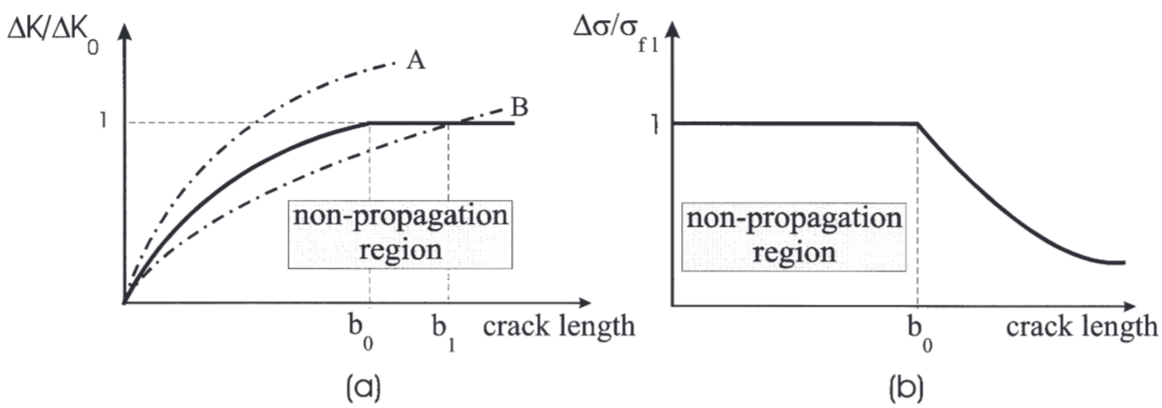


Figure 2-10 K-T diagram (Araújo and Nowell 1999).

The methods discussed so far were based on crack initiation or propagation. Considering the effect of both phases in fretting fatigue life prediction was discussed in (Navarro et al. 2008). This study first compares results using the previously developed methods based on multiaxial stress analysis. Following this, it considers the effect of crack propagation as well and argues that considering both phases is required for fretting fatigue life prediction. This paper presents a new method for considering both phases. In this work, the rate of crack initiation and propagation is compared at different crack lengths (see Figure 2-11). The critical crack length at which crack propagation governs is then determined and the rest of the analysis is done with that crack length and the crack propagation model. Linear elastic fracture mechanics and the Paris-Erdogan law are used for crack propagation calculation in this study. For crack initiation, SWT and FS parameters are used. It is found that the results based on all of the multiaxial stress parameters are relatively close.

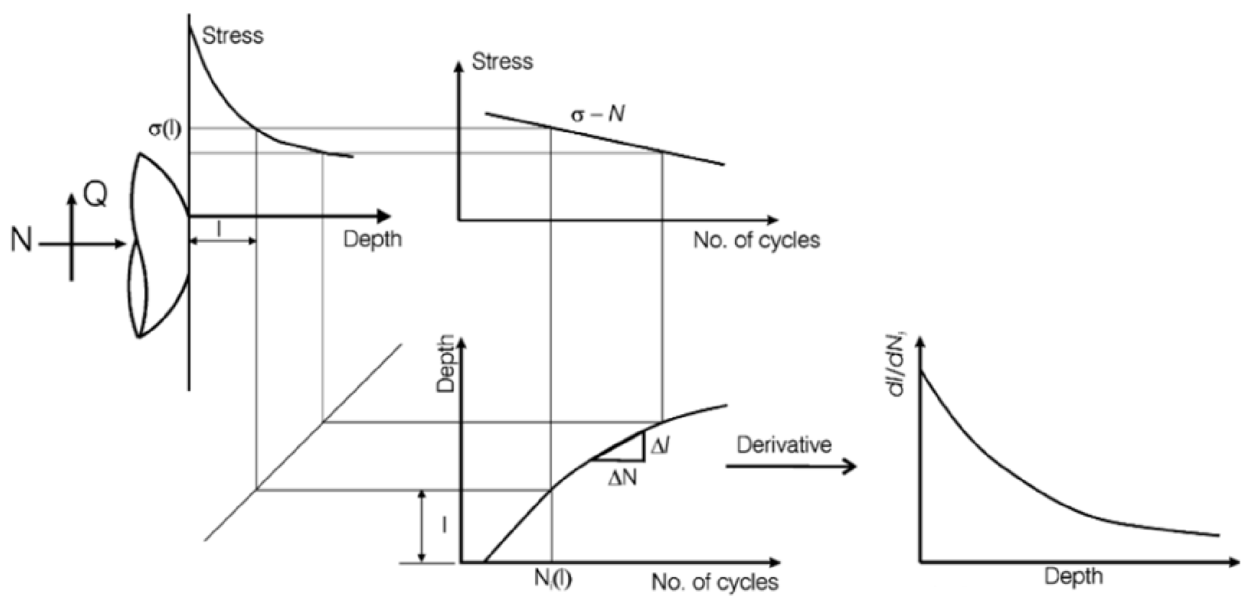


Figure 2-11 Variable crack initiation length concept (Navarro et al. 2008).

Fouvry and Kubiak (2009) employed mixed crack initiation and short crack arrest to predict if a fretting failure occurs. Based on their work, three different regions can be defined (see Figure 2-12): a region in which no crack initiation occurs, another region in which cracks initiate but do not propagate, and another region where cracks initiate and propagate and failure occurs. In this work, multiaxial stress parameters were employed for crack initiation, and the well-known Kitagawa-Takahashi method was employed for short crack arrest evaluation.

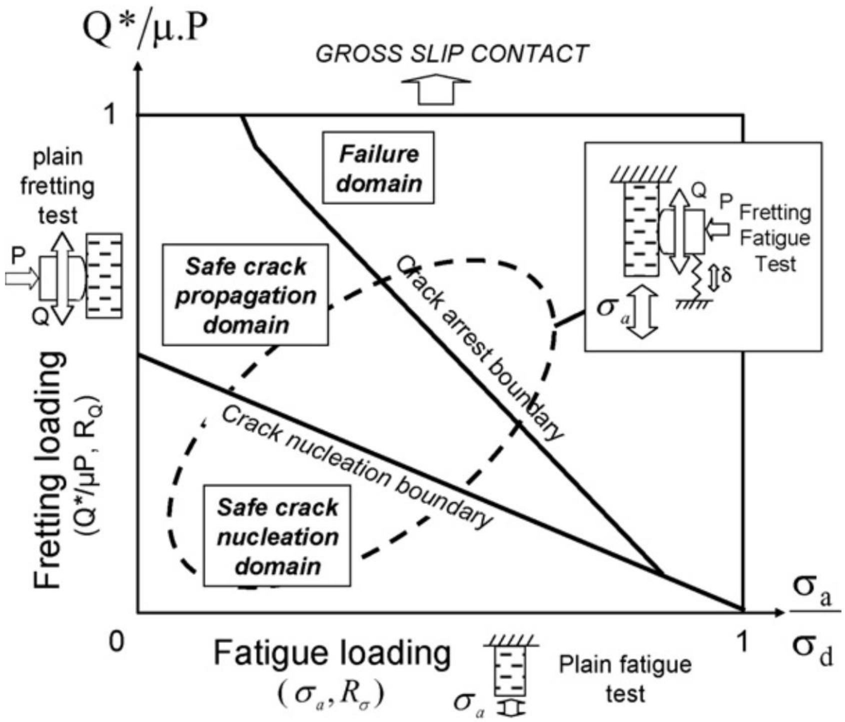


Figure 2-12 Crack nucleation – arrest fretting map (Fouvry and Kubiak 2009).

Asai (2014) identified two stages of crack growth in fretting fatigue problems. Based on this work, cracks initiate at some angle θ_1 to the axis perpendicular to the contact surface (see Figure 2-13). Then a mixed mode regime was seen at a steeper angle. Finally, the cracks continue to propagate in Mode I, perpendicular to the contact surface. Several tests are reported. Most of the crack arrests were seen in Stage 2 shown in Figure 2-13.

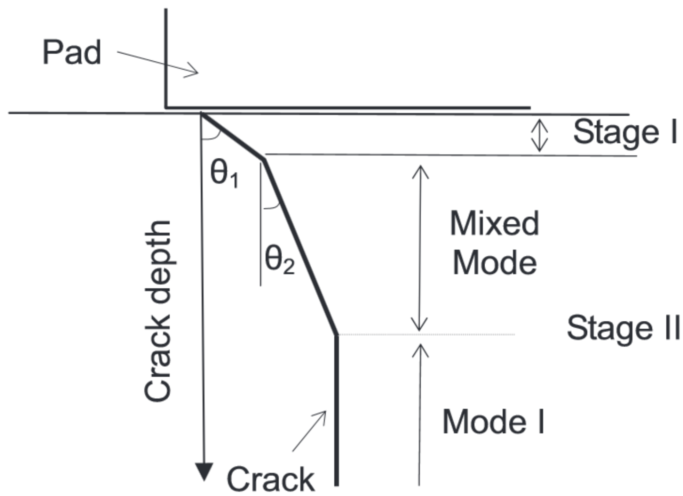


Figure 2-13 Stages of fretting fatigue crack growth (Asai 2014).

2.5 Calculation of critical parameters at the contact points between a cable and a saddle

The critical parameters at the contact points between a cable and a saddle are the contact force, the slip displacement, and the normal stress range. Several studies were performed on this topic by researchers at TU Berlin (Mohareb et al. 2016, Mohareb et al. 2017, Mohareb 2020). The following paragraphs summarize the work performed in these studies.

2.5.1 The axial force of the cable

Due to the frictional loss between a cable and a saddle, the load applied to the cable end is not equal to the axial force of the cable along the saddle. During the loading phase of the cable, the axial force of a cable over a saddle at the central angle θ (defined in Figure 2-14) can be determined based on the (Eytelwein 1808) principle as follows (Mohareb et al. 2017):

$$S_L(\theta) = S_{max} \cdot e^{-\mu \cdot \theta} \qquad 2\text{-}12$$

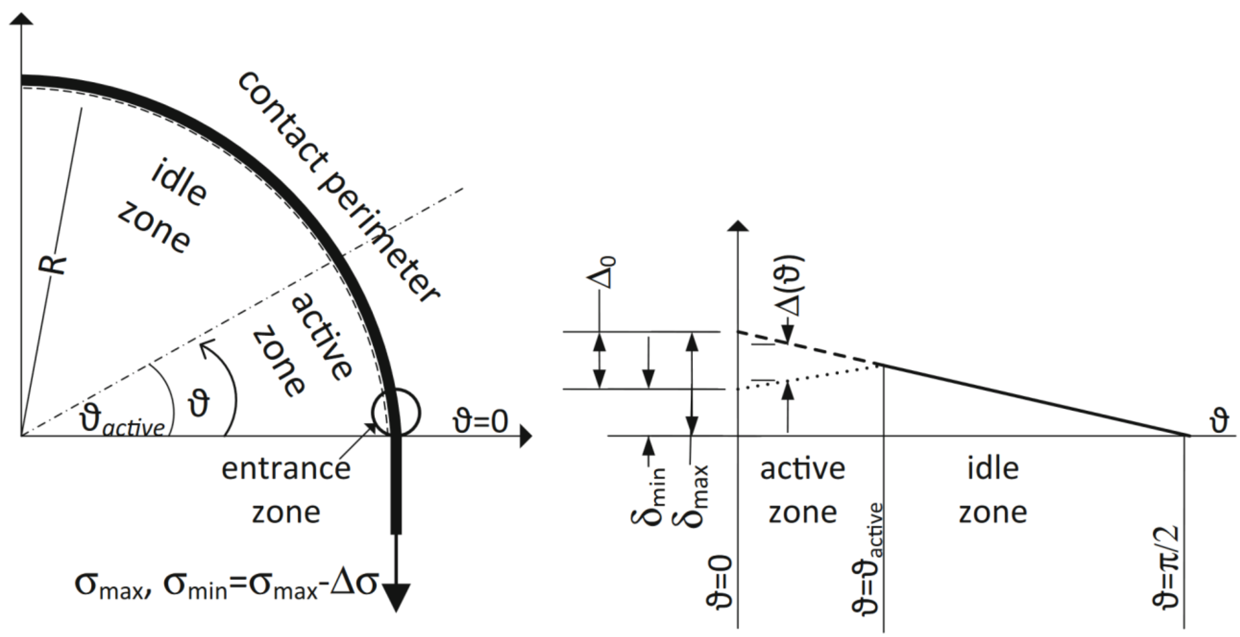
where S_{max} is the applied axial load to the cable during the loading, μ is the COF, and $S_L(\theta)$ is the axial force of the cable at the central angle θ during the loading phase.

During the unloading phase, due to the frictional force, the axial force of the cable is higher along the saddle in comparison with at the end of the cable. In this phase, the axial force can be determined as follows (Mohareb et al. 2017):

$$S_U(\theta) = S_{min} \cdot e^{\mu \cdot \theta} \qquad 2\text{-}13$$

in which S_{min} is the axial force at the cable end and $S_U(\theta)$ is the axial force of the cable at the central angle θ during the unloading phase. It should be noted that the load in the unloading phase cannot be higher than the load in the loading phase. Active and idle zones can be defined for a cable bent over a saddle. There is slip displacement and therefore a stress range in the active region (see Figure 2-14). The border of the two zones, as defined by the angle θ_{active}, can be determined by setting Equation 2-12 equal to Equation 2-13. The simplified equation for calculating θ_{active} is as follows (Mohareb et al. 2017):

$$\theta_{active} = \frac{\ln\left(\sqrt{\frac{S_{max}}{S_{min}}}\right)}{\mu} \qquad \text{2-14}$$



(Note: θ in thesis text = ϑ in this figure.)

Figure 2-14 Active and idle zones for a cable bent over a saddle (Mohareb et al. 2018)

2.5.2 Slip displacement

Mohareb (2020) derived the following equation for the relative displacement between a cable cyclically loaded over a saddle at the central angle θ (See Figure 2-14) in the active region:

$$\Delta(\theta) = \frac{R}{E \cdot \mu \cdot A} \cdot \left(S_{max} \cdot \left(e^{-\mu \cdot \theta} - \sqrt{\frac{S_{min}}{S_{max}}} \right) + S_{min} \cdot \left(e^{\mu \cdot \theta} - \sqrt{\frac{S_{max}}{S_{min}}} \right) \right) \qquad 2\text{-}15$$

where *R* is the radius of the saddle, *E* is the elastic modulus of the cable, *A* is the cross-section area of the cable, μ is the coefficient of friction between the cable and the saddle, S_{max} is the maximum axial force, and S_{min} is the minimum axial force. Looking at this equation, it can be seen that the maximum slip displacement occurs at $\theta = 0$ and can be calculated as follows:

$$\Delta = \frac{R}{E \cdot \mu \cdot A} \cdot \left(S_{max} \cdot \left(1 - \sqrt{\frac{S_{min}}{S_{max}}} \right) + S_{min} \cdot \left(1 - \sqrt{\frac{S_{max}}{S_{min}}} \right) \right) \qquad 2\text{-}16$$

2.5.3 Contact force

The Barlow equation is the main equation used for calculating the contact force in this problem. According to this equation, the contact pressure between a rope over a saddle can be calculated as follows (Mohareb et al. 2017):

$$q = \frac{S}{R} \qquad 2\text{-}17$$

Where q is the contact pressure (Pa), *S* is the axial force (N) of the rope and *R* is the radius (m) of the saddle. For a cable bent over a saddle, due to the twisting angle of the outer wires, the cable is in contact with the saddle at discrete points. The force at each contact point can be calculated using the following equation (Mohareb et al. 2017):

$$F_p = \frac{S}{R} \cdot \frac{l}{n_{out}} \qquad 2\text{-}18$$

where l is the lay length of the cable, and n_{out} is the number of outer wires in a cable. The lay angle is the length required for an outer wire to completely rotate around the central wire (see Figure 2-15).

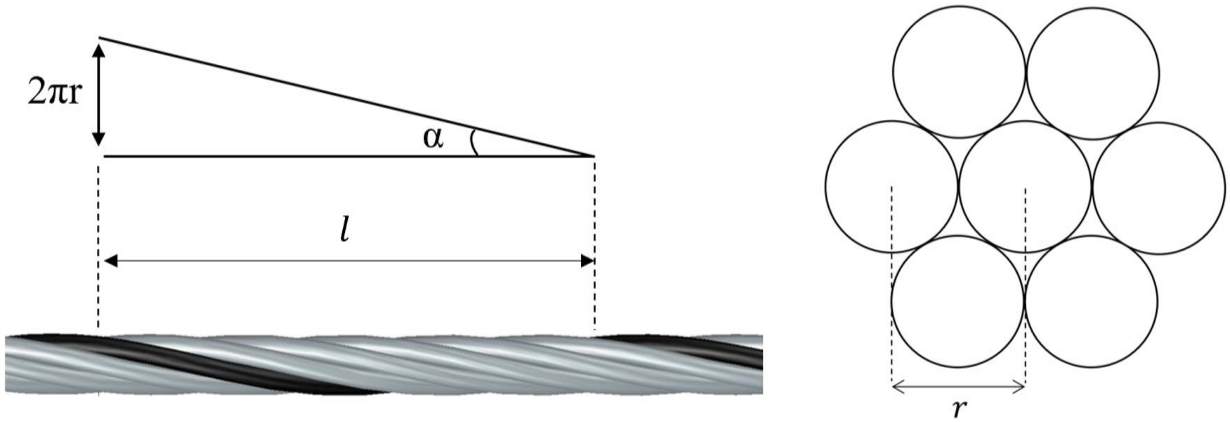


Figure 2-15 Geometric parameters of a cable.

One issue with using these equations is they assume there is uniform contact pressure between the cable and saddle over the entire contact surface. However, several studies show there is a non-uniform region at the point where the cable first meets the saddle, and the results of these equations are not accurate at the first few contact points (Molkow 1982, Wiek 1982, Feyrer 2007).

2.5.4 Stress range

Another parameter that affects the fretting fatigue life of the cable is the remote or normal stress range of the wire. This stress range consists of the normal stress due to the axial force of the cable and the stress caused by bending the cable over the saddle. The normal stress due to the axial force of the cable can be calculated using the following equation (Mohareb 2020):

$$\sigma_{T_i} = \frac{\frac{E_i cos^2(\alpha_i)}{1 + \nu \cdot \sin^2(\alpha_i)}}{\sum_{j=1}^{n} \frac{E_j A_j \cos^3(\alpha_j)}{1 + \nu \cdot \sin^2(\alpha_i)}} S \qquad \text{2-19}$$

where E_j and A_j are the elastic modulus and area of each wire, ν is the Poisson's ratio, α_j is the lay angle of each wire, and S is the axial load applied to the cable.

The bending stress due to the bending of the cable over the saddle can be calculated using the difference between the curvature of the outer wire before and after bending the cable over the saddle. The curvature of the wire before bending can be calculated using the following equation (Feyrer 2007, Mohareb et al. 2020):

$$\frac{1}{\rho} = \frac{\sin^2(\alpha)}{r_w} \qquad 2\text{-}20$$

where r_w is the helix radius of the cable shown in Figure 2-15 and α is the lay angle of the cable. Figure 2-16 shows the geometry parameters of an outer wire of a cable bent over a saddle. The curvature after twisting can be calculated using the following equation (Feyrer 2007):

$$\frac{1}{\rho} = \frac{(x'^2 + y'^2 + z'^2) \cdot (x"^2 + y"^2 + z"^2) - (x' \cdot x" + y' \cdot y" + z' \cdot z")}{(x'^2 + y'^2 + z'^2)^3} \qquad 2\text{-}21$$

The parametric equation of the outer wire is required to determine the parameters required in Equation 2-21. The parametric equation of a cable outer wire bent over a saddle can be written as follows (Feyrer 2007):

$$x = -r_w \cdot \sin(\varphi) \qquad 2\text{-}22$$

$$y = \frac{D}{2} \cdot \cos(\theta) + r_w \cdot \cos(\varphi) \cdot \cos(\theta) \qquad 2\text{-}23$$

$$z = \frac{D}{2} \cdot \sin(\theta) + r_w \cdot \cos(\varphi) \cdot \sin(\theta) \qquad 2\text{-}24$$

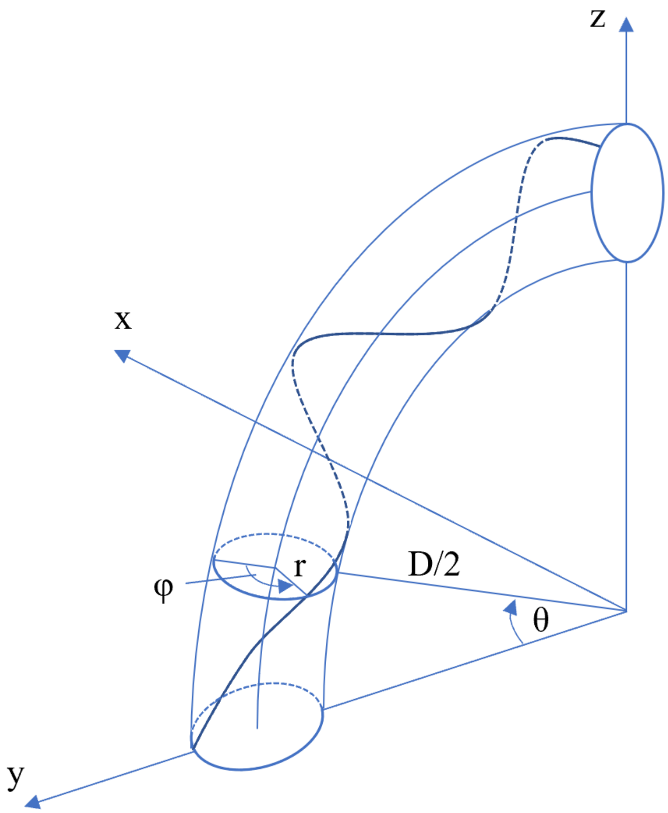


Figure 2-16 Geometric parameters for the outer wire of a cable bent over a saddle (based on Schiffner (1986)).

A simpler equation for determining the curvature is presented in (Wiek 1976, Hobbs and Nabijou 1995). Based on these studies, the curvature of an outer wire of a bent cable can be determined using the following equation:

$$\frac{1}{\rho} = \frac{(G-H)^{\frac{1}{2}}}{Q} \tag{2-25}$$

where:

$$G = 1 + c \cdot \cot^2(\alpha)\{4c + 2cos(\varphi) - 2c \cdot \cos^2(\varphi) + c \cdot \cot^2(\alpha)(1 + c \cdot \cos(\varphi))^2 \tag{2-26}$$

$$H = \frac{c^2 \cdot \cot^4(\alpha) \cdot \sin^2(\varphi)(1 + c \cdot \cos(\varphi))^2}{1 + \cot^2(\alpha)(1 + c \cdot \cos(\varphi))^2} \tag{2-27}$$

$$Q = r\{1 + \cot^2(\alpha)(1 + c \cdot \cos(\varphi))^2\} \tag{2-28}$$

where r is the helix radius of the wire in the strand and c is equal to the ratio of helix radius to the saddle radius (r/R).

The bending stress of the outer wire can be calculated using the following equation:

$$\sigma_b = r \cdot E \cdot (\frac{1}{\rho_1} - \frac{1}{\rho_o})$$

where $\frac{1}{\rho_1}$ is the curvature of the outer wire after bending the cable over the saddle, and $\frac{1}{\rho_o}$ is the curvature of the outer wire when the cable is over a straight surface.

2.6 Fretting fatigue tests of high-strength steel cables in air and corrosive environments

This section summarizes several studies on fretting fatigue tests of bridge cables. It should be noted that these works study the “interwire” fretting fatigue performance of the cables, and not fretting fatigue between a cable and a saddle system.

Perier et al. (2009) evaluated the interwire fretting fatigue behaviour of bridge cables considering the effect of corrosion. In this study, wire specimens were in contact with other wires in a solution of NaCl. Figure 2-17 (a and b) shows the test setup. Steel wires with an ultimate tensile strength of 1860 MPa were used in this study. Several tests were done to evaluate the effect of lubrication and galvanization on the fretting fatigue life of the wires. The fretting fatigue tests were done at a mean stress of 600 MPa. Based on the results of this study, the fretting fatigue endurance limit was found to be 250 MPa for lubricated, 170 MPa for galvanized wires, and 100 MPa for bright wires. Figure 2-17 (c) shows the fretting fatigue test results of this study. Figure 2-17 (d and e) shows the fracture surface for tests performed in air and in NaCl solution respectively.

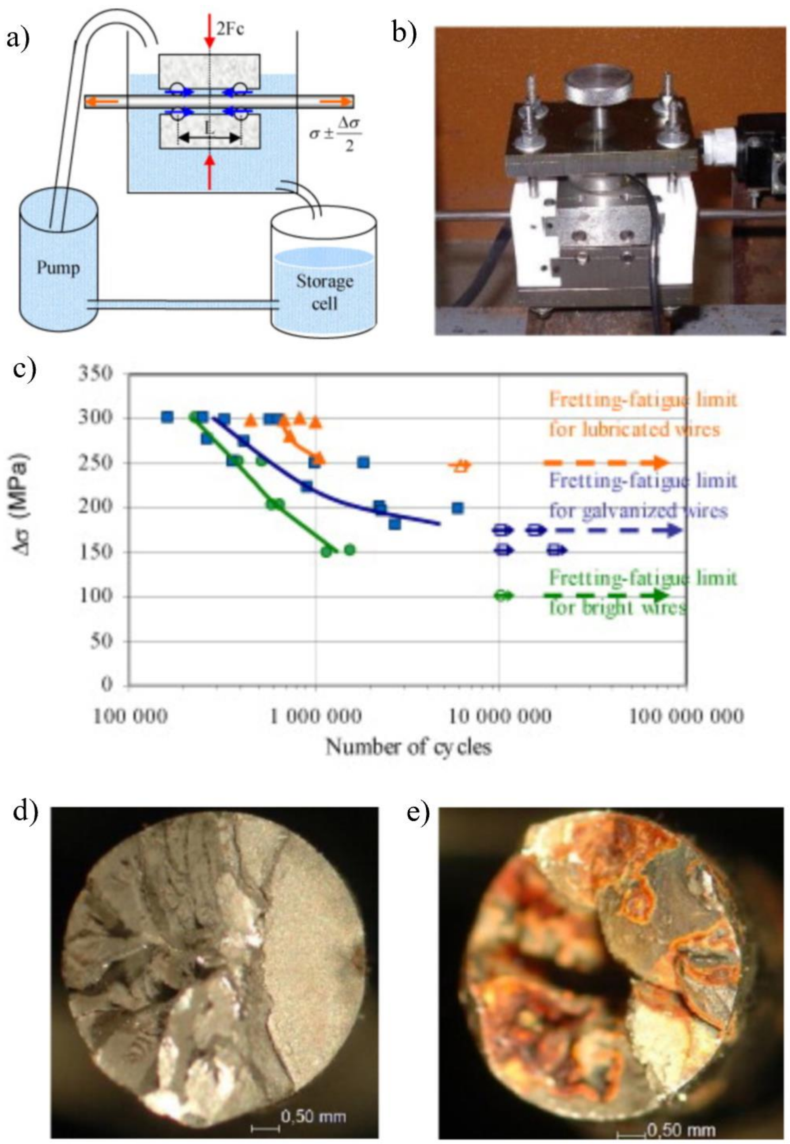


Figure 2-17 Interwire fretting fatigue tests presented in Perier et al. (2009), schematic view of test setup (a), test setup (b), fretting fatigue test results (c), failed wire in air (d) and corrosive environment (e).

Winkler et al. (2015) evaluated the fretting fatigue behaviour of high-strength steel strands under bending loading. Several tests were performed and the interwire relative displacement was measured using the DIC technique. Interwire movements ranging from 30 to 55 µm were recorded. The maximum fretting fatigue life was at a relative displacement of 30 µm (~ 3 million cycles), and the minimum fatigue life was at 55 µm (~ 70 thousand cycles).

Guo et al. (2020) evaluated the interwire fretting fatigue behaviour of bridge cables. A schematic view of the test concept and setup is shown in Figure 2-18 (a, b, and c). In this study, fretting fatigue tests were performed using three displacement ranges of 0.44, 0.66, and 0.88 mm and two contact forces of 60 and 120 N. Based on the results of these tests, the fretting fatigue life decreases with an increase in contact force and displacement range (see Figure 2-18 (d)). However, it should be noted that the higher displacement range is associated with a higher stress range. Therefore, it is not certain that the decrease in fatigue life is only due to the increase in slip displacement. In this paper, wear scars on the wires were measured during the tests using a white light interferometer.

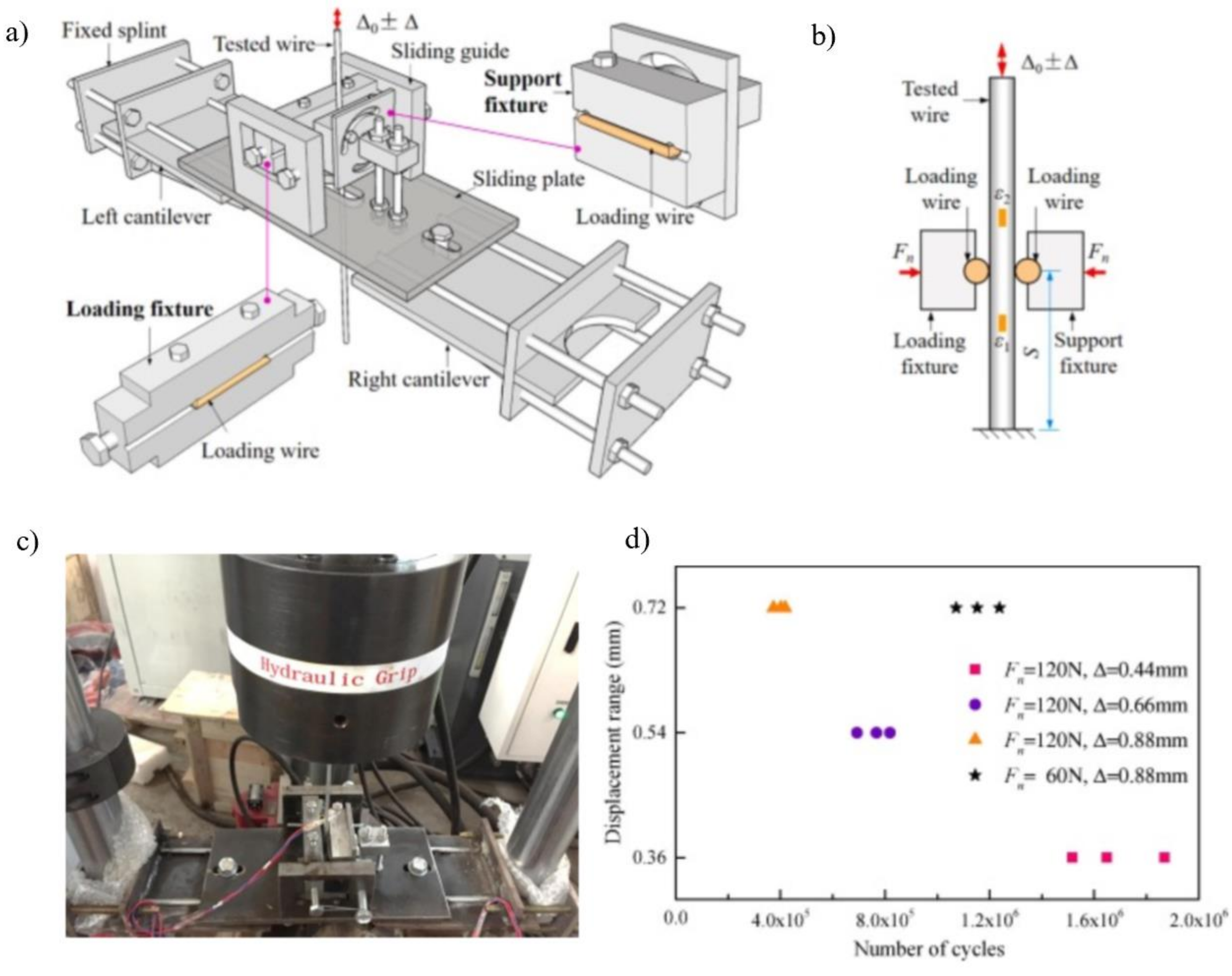


Figure 2-18 Interwire fretting fatigue tests presented in Guo et al. (2020).

Liu et al. (2020) employed NaCl solution and an accelerated corrosion assembly to evaluate the interwire fretting fatigue behaviour of bridge stay cables in corrosive environments. It seems that the setup previously used in Guo et al. (2020) was adapted to include an accelerated corrosion assembly. Several current intensities were employed to evaluate the effect of corrosion on fretting fatigue life. It was shown that the fretting fatigue life decreases with an increase in the current intensity or in other words the corrosion severity.

2.7 Studies on saddle systems at TU Berlin

A series of fretting fatigue tests of saddle systems was undertaken at TU Berlin. First, a full-scale saddle system as shown in Figure 2-19 was tested to evaluate the fretting fatigue performance of cables (Schlaich et al. 2010). Fifty-five strands were used in these tests. The tests were performed according to the recommendation of (fib 2005). According to these standards, the cables were cycled at maximum stress of 45% GUTS, guaranteed ultimate tensile strength, with a stress range of 200 MPa. Following the fatigue tests, the strands were loaded statically to failure. The first strand failed at 96.7% GUTS, and the average tensile capacity of the strands was 99% GUTS. According to (fib 2005), the tensile capacity should be at least 95% GUTS after the fatigue tests. Therefore, the system passed the requirements of (fib 2005).

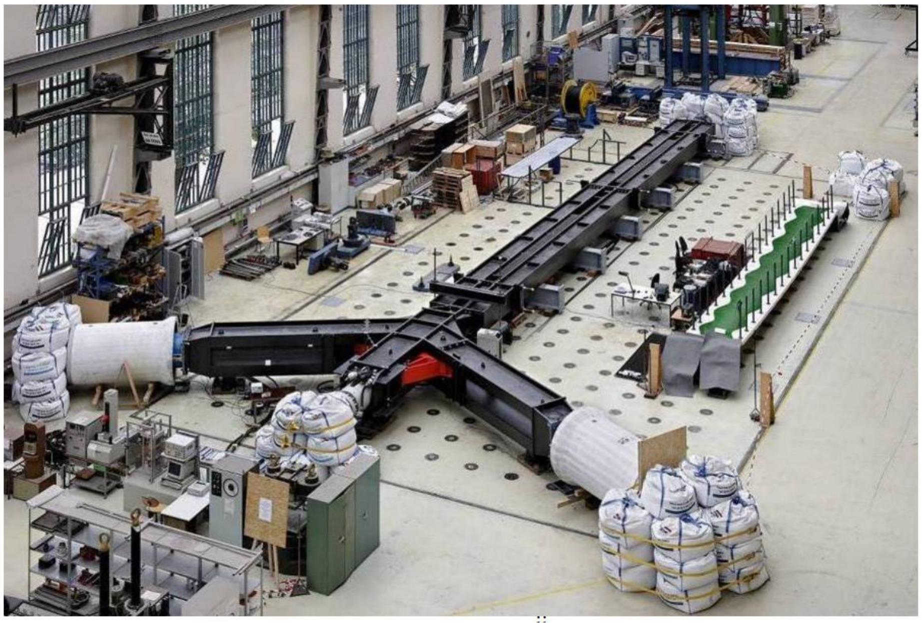


Figure 2-19 Full-scale fretting fatigue tests of saddle systems at TU Berlin (Schlaich et al. 2010).

The full-scale tests were found to be costly and time-consuming. Therefore, a reduced-scale test setup with smaller dimensions was designed and used for further studies on this topic. Figure 2-20 shows this test setup to evaluate the fretting fatigue behaviour of cables in saddle systems. This setup consisted of a steel saddle, which enabled tests to be done at three saddle radii of 500, 1000, and 1500 mm. Three strands can be tested at each radius simultaneously. The load is applied using a single actuator at the center and is monitored at the ends of the cables.

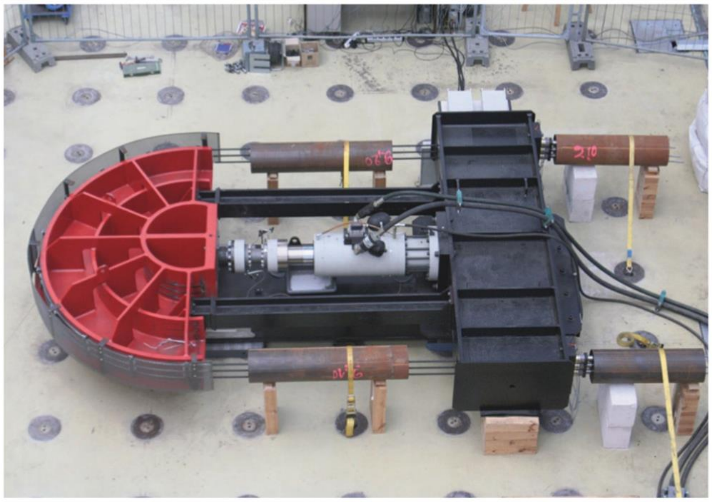


Figure 2-20 Reduced-scale fretting fatigue test at TU Berlin (Schlaich et al. 2012).

In this study, two tests at saddle radius of 1000 mm, and one test each at radii of 500 and 1500 mm were done. These tests were again performed according to the recommendation of (fib 2005). The cables were cycled at a maximum stress range of 45% GUTS with a stress range of 200 MPa. The results of these tests are summarized in Figure 2-21. Interestingly, the shortest fatigue life occurred at the 1000 mm radius. The fretting fatigue life was higher at the saddle radius of 500 mm, and no failure was observed at the saddle radius of 1500 mm up to 2 million cycles. It should be noted that different critical parameters, specifically contact force and slip displacement, work against each other when the saddle radius increases. A smaller saddle radius is associated with a higher contact force and a smaller slip displacement.

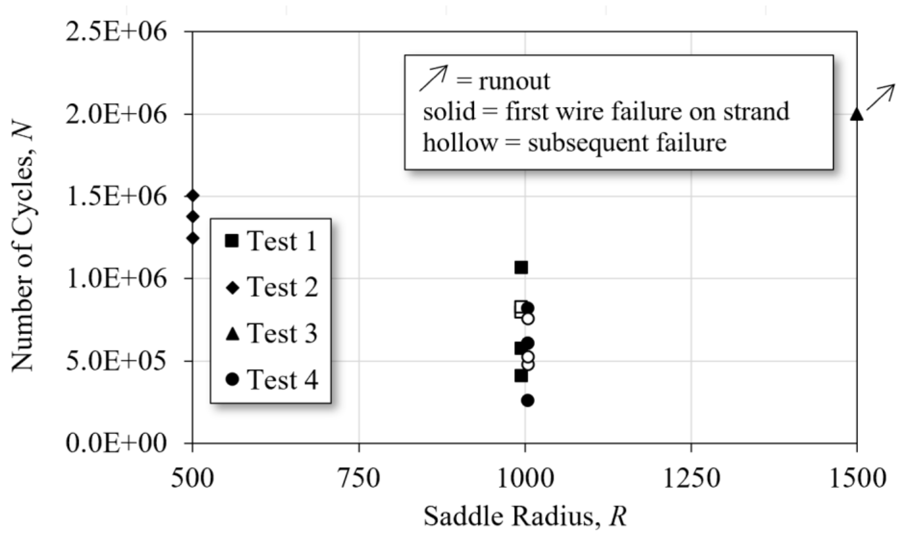


Figure 2-21 Fretting fatigue test results at TU Berlin (Schlaich et al. 2016).

Following this experimental work, a fretting fatigue analysis of this system was performed and presented at the IABSE conference (Mohareb et al. 2017). In this study, the critical parameters at the contact points were determined based on the methods previously discussed in Section 2.5 and were used as input parameters for an interface model of a cable over a saddle (see Figure 2-22). This study employed multiaxial stress analysis based on the SWT parameter to determine the fretting fatigue life of the cables. It also discusses the possibility of using the approach proposed by Ding et al. (2010) to consider the effect of wear. However, given that wear constants for the method presented in Ding et al. (2010) cannot be easily found in the literature, a trial-and-error approach was used to show how this parameter can improve fretting fatigue life predictions.

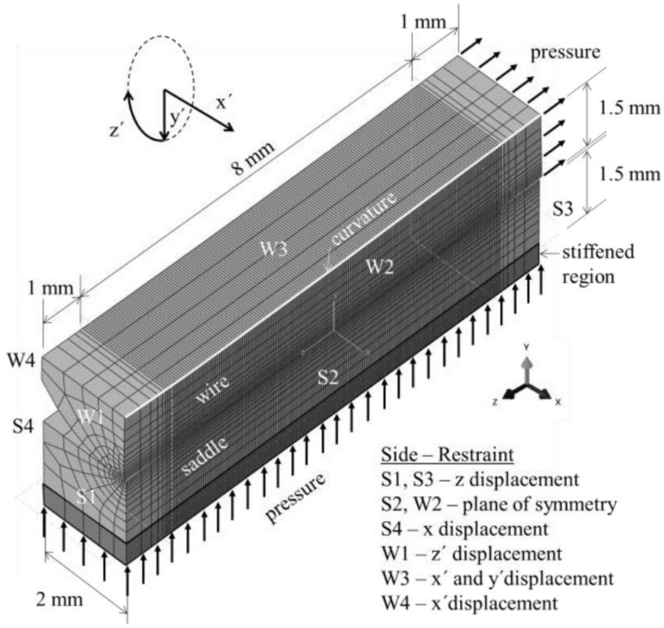


Figure 2-22 Interface model of a cable over a saddle (Walbridge et al. 2017).

2.8 Fretting fatigue test setups

Several fretting fatigue test setups varying in scale have been employed in different engineering fields to evaluate the fretting fatigue behaviour of components. In this part, three different test setups employed at different universities are described and discussed.

2.8.1 Fretting fatigue test setup at Ghent University

Figure 2-23 shows a schematic view of a fretting fatigue test setup developed by researchers at Ghent University in Belgium (Hojjati-Talemi 2014). In this setup, a 100 kN hydraulic cylinder is used to apply cyclic loads to dog bone specimens. A fretting fatigue fixture is used to hold the pads and apply the contact force to the specimens. The pads are fixed to two elastic frames. These frames are designed to be horizontally flexible. A 10 kN servo-hydraulic actuator installed on a C-beam is used to apply the lateral contact force. The C-beam is installed on low friction material, allowing it to move freely in the horizontal direction to ensure that the normal forces are equal and there is no bending in the specimens.

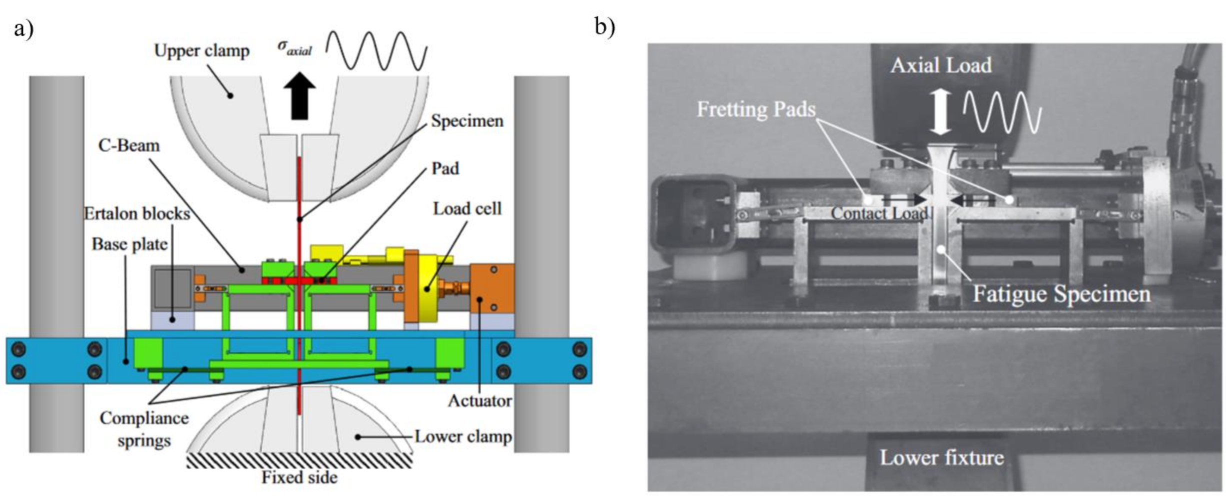


Figure 2-23 Fretting fatigue test setup developed at Ghent University (Hojjati-Talemi 2014).

2.8.2 Fretting fatigue test setup at Purdue University

Figure 2-24 shows a schematic view of a test setup developed at Purdue University (Gean 2008, Srinivasan et al. 2009). The cyclic load is applied by a hydraulic cylinder to dog bone specimens. The cyclic load is applied at the bottom cross-head and the top cross-head is fixed. There are two load cells at the bottom and at the top crosshead to measure the difference between the cyclic load at the top and bottom of the specimens. The pads are held with a fretting fatigue fixture that is shown in Figure 2-24. The contact force is applied by two hydraulic actuators on the left and right sides of the pad holders. Washer load cells have been used on the through-rods to measure the applied contact force. As with the Ghent setup, the stiffness of the fretting fatigue chassis is low in the horizontal direction but high in the vertical direction to transfer the high tangential loads because of friction in the contact area. The normal load transfer rate is measured to be greater than 95%.

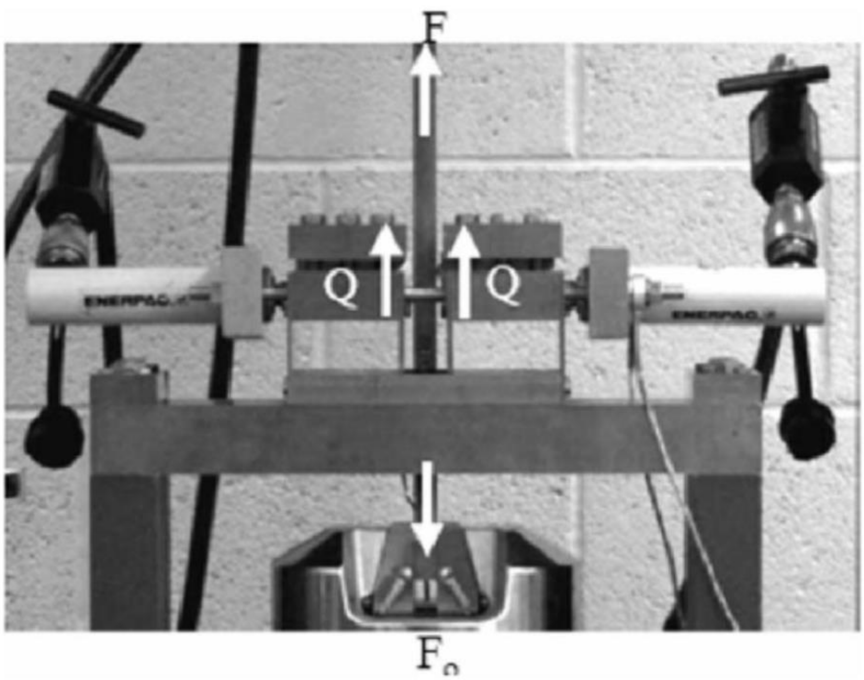


Figure 2-24 Fretting fatigue test setup developed at Purdue University (Srinivasan et al. 2009).

2.8.3 Fretting fatigue test setup at University of Brasilia

Figure 2-25 shows a newer fretting fatigue test setup developed recently at the University of Brasilia (Matos et al. 2020, Araújo et al. 2020). It employs similar concepts to the two setups described in the previous sections. It is noteworthy for having been developed specifically to study the problem of inter-wire fretting in cables.

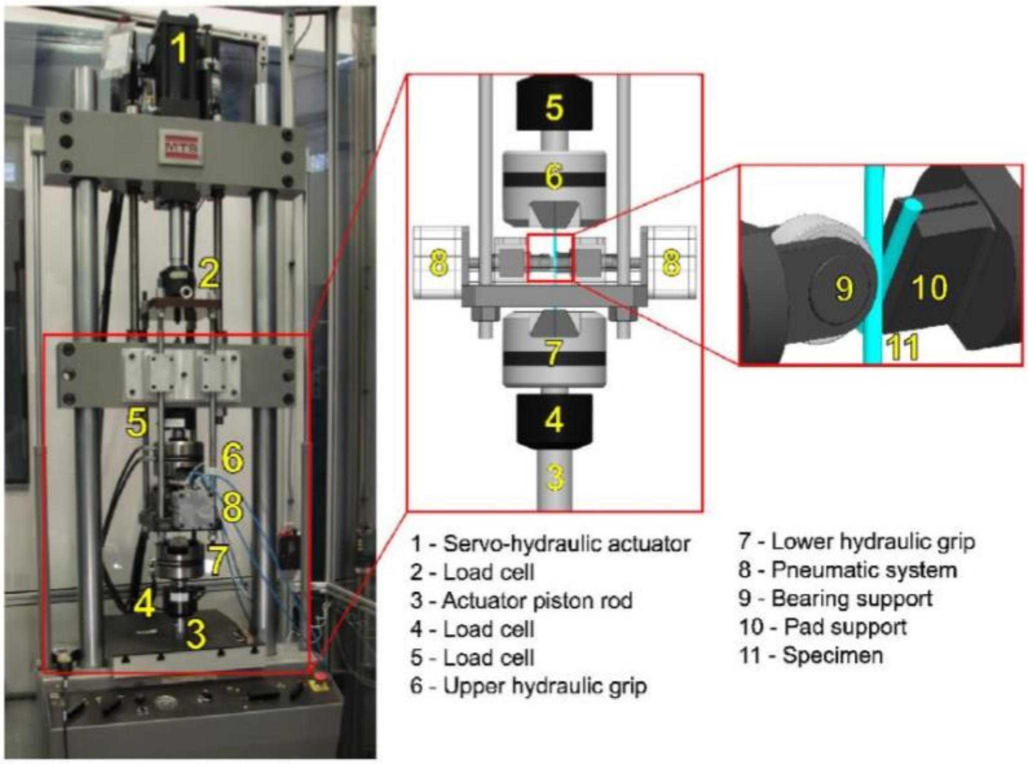


Figure 2-25 Fretting fatigue test setup developed at the University of Brasilia (Araújo et al. 2020).

2.9 Summary

This chapter first went over the fretting fatigue phenomenon and the effective parameters that affect the fretting fatigue behaviour of contacting components. Based on the literature, the most important parameters are contact force, slip displacement, normal stress, and surface material properties including coefficient of friction between the contacting components, surface hardness, etc. Corrosion also was found to be another factor that can play a role in fretting fatigue failures and needs to be considered when the components are used in corrosive environments. This chapter then went over different crack initiation and crack propagation approaches to determine the fretting fatigue life of components. High stress gradients seem to be a major challenge when initiation criteria are being used. Short crack arrest and size effects add another level of complexity to the fretting fatigue problems. Different approaches including averaging over a volume/surface/line have been employed to overcome these issues. Theory of critical distance with and without mesh control and considering a coarse mesh size are other approaches to consider the effect of high stress gradients in fretting fatigue problems. It seems that some sort of calibration is required with all of these methods and the efforts to relate the parameters of these methods to material properties or microstructure (e.g., grain size) have not yet been completely successful. Crack propagation approaches using linear elastic fracture mechanics also have been employed in several studies to evaluate the fretting fatigue problems. These approaches have been used with and without crack initiation criteria. When it comes to fretting fatigue analysis of cable wires, very little can be found in the literature on their behaviour in contact with saddles. However, several works can be found on interwire fretting fatigue of mine ropes and bridge cables. The works on fretting fatigue analysis of cables at saddle supports are limited to full-scale or large-scale tests which are time-consuming and costly. Given the fact that the fretting fatigue failures of bridge cables are a major design consideration for saddle systems, a number of studies are required on this topic to evaluate the fretting fatigue performance of cables at saddle supports both numerically and experimentally.

3. Fretting Fatigue Analysis of Bridge Stay Cables at Saddle Supports

As discussed in the previous chapter, several methods can be used to determine the fretting fatigue life of components. As the radius of the cable wires is typically small, most of the fatigue life can be attributed to the crack initiation phase. Therefore, employing a crack initiation criterion was found to be a reasonable assumption. In this work, a multiaxial stress analysis approach employing the SWT parameter is employed. This chapter starts with presenting the framework used to determine the fretting fatigue life of bridge stay cables at saddle supports. Each part of the framework is then discussed in more detail. To show how this approach works, the parameters and geometry of the fretting fatigue tests at TU Berlin (Schlaich et al. 2016) are used as an example.

3.1 Deterministic framework for evaluating fretting fatigue life

The procedure used herein to evaluate the fretting fatigue life of bridge stay cables is as follows:

1. Calculate critical parameters (slip displacements, contact forces, and normal stresses) in the contact area, given the saddle geometry and the load.
2. Determine Coffin-Manson material parameters (σ'_f, ε'_f, b, c) through fatigue tests on the wire material or using an empirical model from the literature.
3. Use FE analysis and the critical planes method to find the SWT parameter.
4. Determine fatigue life using the peak value of the SWT parameter obtained in Step 3 and Coffin-Manson material parameters from Step 2.

This procedure is shown graphically in Figure 3-1, and more details concerning each part of this framework are presented in the following sections.

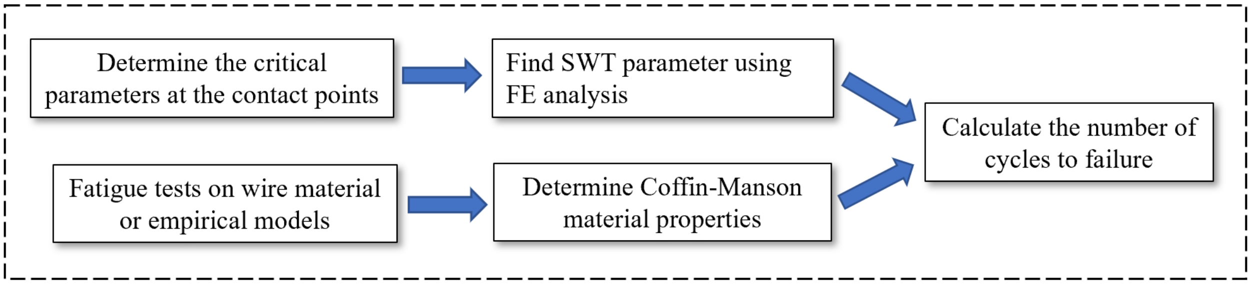


Figure 3-1 Deterministic framework for evaluating fretting fatigue life.

3.2 The geometry of the saddle and cable

Figure 3-2 shows a schematic view of a cable over a saddle. Discrete contact points between the cable and the saddle can be seen in this figure. The saddle radius is the main geometric parameter of the saddle. The cable is made of a central wire and six wires twisting around the central wire. One of the main properties of a cable is the lay length, which is equal to the length required for an outer wire to completely rotate around the central wire (see Figure 3-3). This property of the cable can also be reported in terms of an angle, called lay angle, determined as follows:

$$\alpha = \frac{2\pi r}{h} \qquad \text{3-1}$$

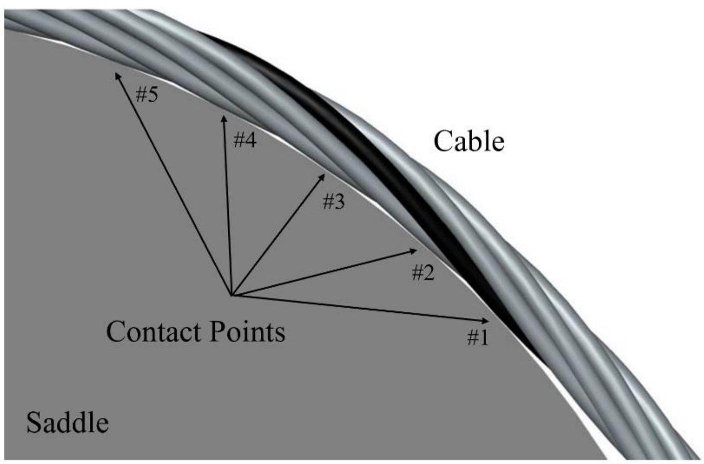


Figure 3-2 Schematic view of a cable over a saddle.

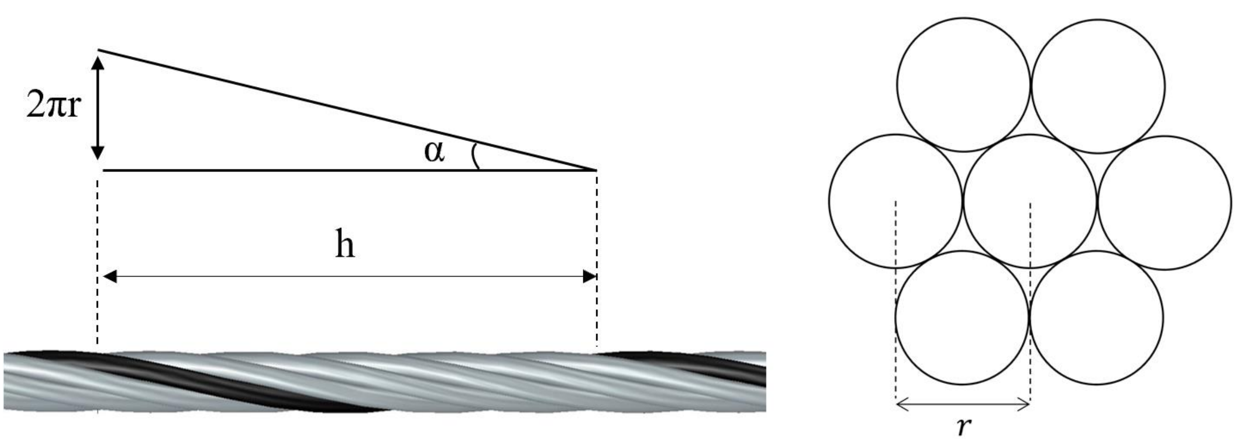


Figure 3-3 Lay angle and lay length of a cable.

3.3 Determination of critical parameters at contact points

The critical parameters at the contact points, required to determine the fretting fatigue life of the bridge stay cables, are contact force, normal stress, and slip displacement. These critical parameters are the inputs for the FE model of a single contact point between the cable and the saddle. Details on calculating these parameters can be found in Mohareb et al. (2016) and Mohareb et al. (2017). These studies are summarized in Chapter 2. In these studies, closed-form equations are established for each parameter, considering the mechanics of the problem of a strand with non-zero flexural stiffness draped over a cable and subjected to a tensile force. The flexural stiffness of the strand is taken into consideration in this analysis, as well as the effects of friction between the saddle and the cable. The critical parameters are functions of the loading, material properties, coefficient of friction (COF), and geometry of the cable and the saddle. There are two possible issues with the closed-form equations. First, these equations are based on the Barlow equation and assume a continuous contact between the saddle and the cable. However, the cable and the saddle are in contact at discrete points (as shown in Figure 3-2). Second, several works have shown that there is a nonuniform region for the contact pressure at the region where the cable first meets the saddle (Wiek 1982, Molkow 1983, Feyrer 2007). The effect of this nonuniform region on the critical parameters cannot be simply considered with the closed-form equations. Therefore, it is required to evaluate the accuracy of the closed-form equations in the nonuniform region at the first contact points. Mohareb et al. (2020) employed a 2D FE model to evaluate the accuracy of the closed-form equations for the contact force. In this FE model, the cable was modelled as a band and the saddle was modelled as rigid arches. The band had the same bending stiffness as the bending stiffness of the cable. This model was used for contact force evaluation at the contact points. The area of the band, however, was not equal to the area of the cable. The area of the cable is one of the parameters that affect the slip displacement at the contact points. Therefore, this same model could not be used for slip displacement evaluation.

For the current study, all the methods based on the closed-form equations were implemented in an Excel spreadsheet. Also, a new 2D FE model with a band with a similar area and bending stiffness of the cable was used to evaluate contact forces and slip displacements. The main difference between this model and the model previously developed in Mohareb et al. (2020) is that the band has the same area as the area of the cable in this model. Therefore, this model could be used for slip displacement evaluation. In the following sections, first, the 2D FE model is discussed. Then the results of the 2D FE model are compared with closed-form equations in the literature.

3.3.1 2D FE model of a cable over the saddle

The FE program ABAQUS was used to formulate this problem. Figure 3-4 shows the 2D FE model of the cable over the saddle. The cable was modelled as a band with a cross-section that has the same area and bending stiffness as the cable. To get discrete contact points instead of continuous contact, the saddle was modelled as a series of rigid arches. The radius of these arches was equal to the curvature of the outer wire of the cable when the cable is on a straight surface (R = 230.4 mm). Three different saddle radii of 500, 1000, 1500 mm were modelled, similar to the saddle radii of the tests at TU Berlin. The length of the band was three times the radius of the saddle. The band was fixed at one end above the center of the saddle and the load was applied to the band at the other end. Four node plane stress quadrilateral (CPS4R) elements with an approximate size of 1 × 1 mm were used for meshing the band. Hard contact along with a penalty method was used to model the normal contact behaviour. Different coefficients of friction, COFs, namely: 0.2, 0.4, 0.6, and 0.8 were used to evaluate the effect of the COF on the results. The friction was imposed with the penalty algorithm in ABAQUS. Based on the design stress specified in (fib 2005), the cable was cycled between a maximum load of 126 kN (837 MPa) and a minimum load of 96 kN (637 MPa). This load was applied to the band in two steps and was distributed between the nodes at the end of the band. Given the changes in the geometry and the COF, a Python script was written to make the FE models, run them and save the results.

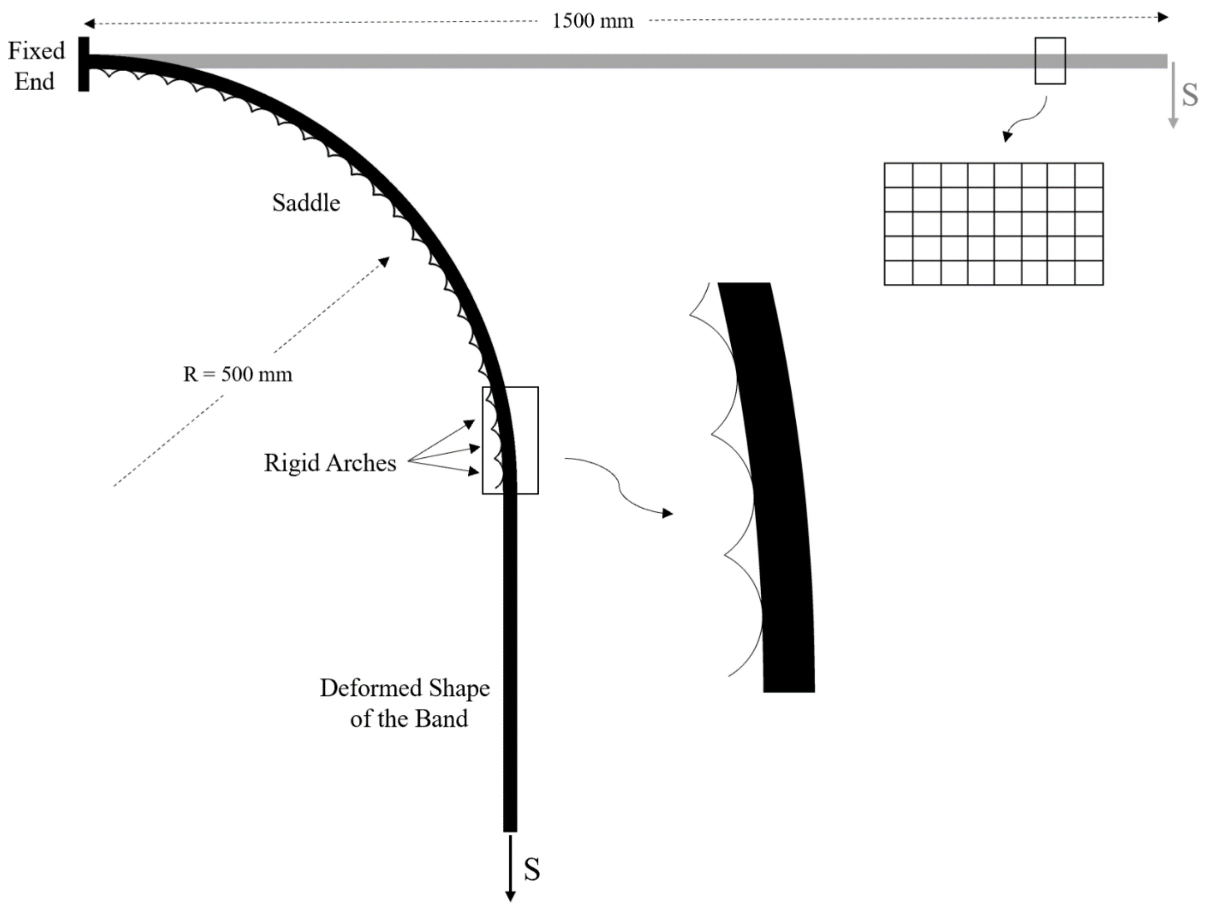


Figure 3-4 2D FE model of a cable bent over a saddle.

3.3.2 Slip displacements

Slip displacement at the central angle θ can be calculated as follows (Mohareb 2020):

$$\Delta(\theta) = \frac{R}{E \cdot \mu \cdot A} \cdot \left(S_{max} \cdot \left(e^{-\mu \cdot \theta} - \sqrt{\frac{S_{min}}{S_{max}}} \right) + S_{min} \cdot \left(e^{\mu \cdot \theta} - \sqrt{\frac{S_{max}}{S_{min}}} \right) \right) \qquad \text{3-2}$$

where R is the radius of the saddle, μ is the COF between the cable and the saddle, E is the elastic modulus of the cable, A is the area of the cable. S_{max} is the maximum cable axial force, 126 kN, and S_{min} is the minimum cable axial force, 96 kN. It should be noted that the saddle can be divided into two parts: an active region and a non-active region where there is no displacement. θ_{active} can be determined as follows (Mohareb 2020):

$$\theta_{active} = \frac{\ln\left(\sqrt{\frac{S_{max}}{S_{min}}}\right)}{\mu} \quad \text{3-3}$$

The results of Equation 3-2 are compared with the FE model results for three different saddle radii of 500, 1000, 1500 mm and COFs of 0.2, 0.4, 0.6, 0.8 in Figure 3-5 to Figure 3-8. First, it can be seen that the number of points in the active region, where slip occurs, increases with an increase in the saddle radius or a decrease in the COF as the frictional force decreases in these cases. The results of the FE model are close to the results of Equation 3-2, especially for the cases with high saddle radius and low COF. However, the difference gets higher for the cases with higher COF and lower saddle radii. The maximum difference in slip displacement results was 0.02 mm. Typically, the saddle radius of real bridges is greater than 500 mm. The results of the analytical equations and the FE model were closer for saddle radii in this range. All in all, this comparison shows that employing Equation 3-2 is conservative (in the sense that it slightly overestimates slip) for the first contact points, where fretting fatigue failure occurs.

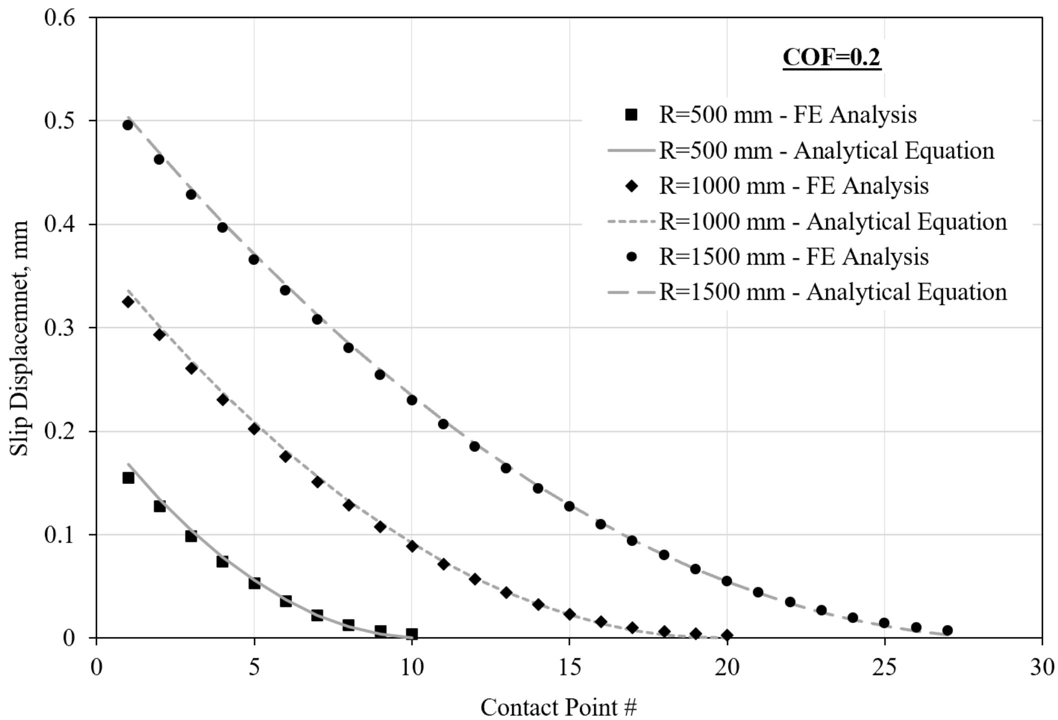


Figure 3-5 Slip displacement results for a COF of 0.2.

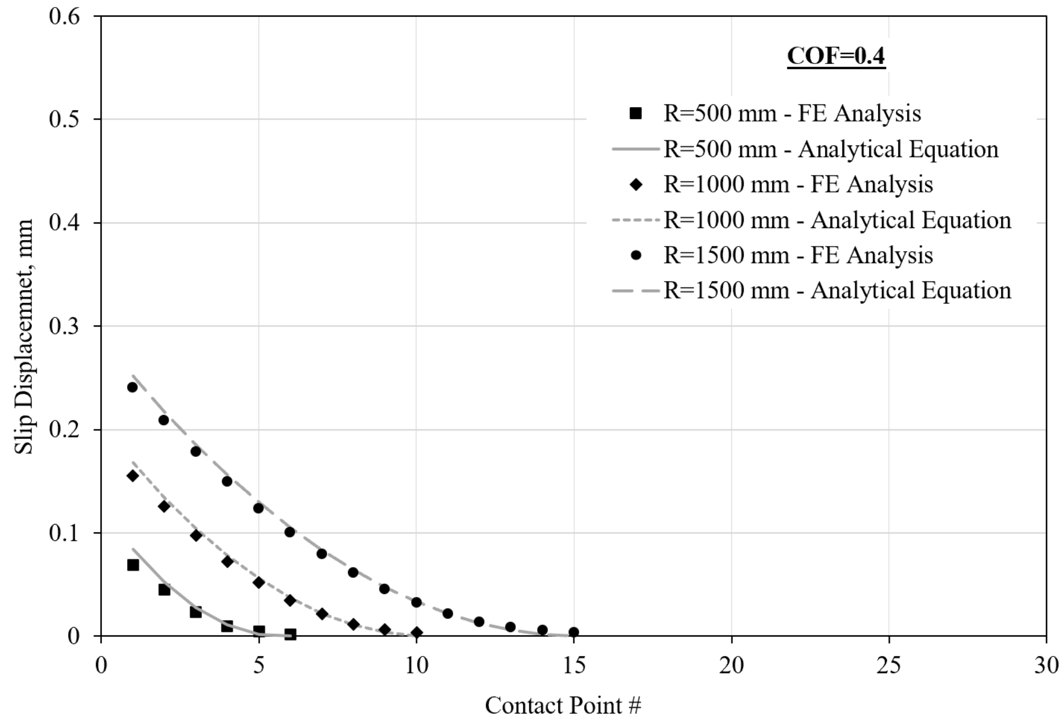


Figure 3-6 Slip displacement results for a COF of 0.4.

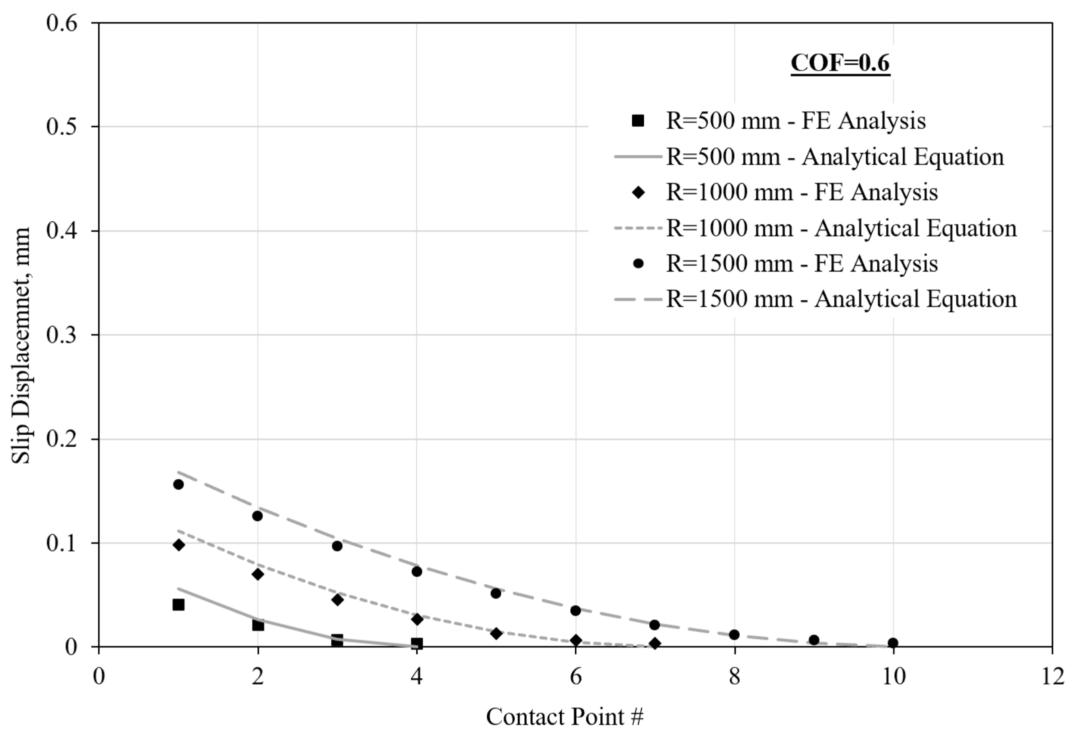


Figure 3-7 Slip displacement results for a COF of 0.6.

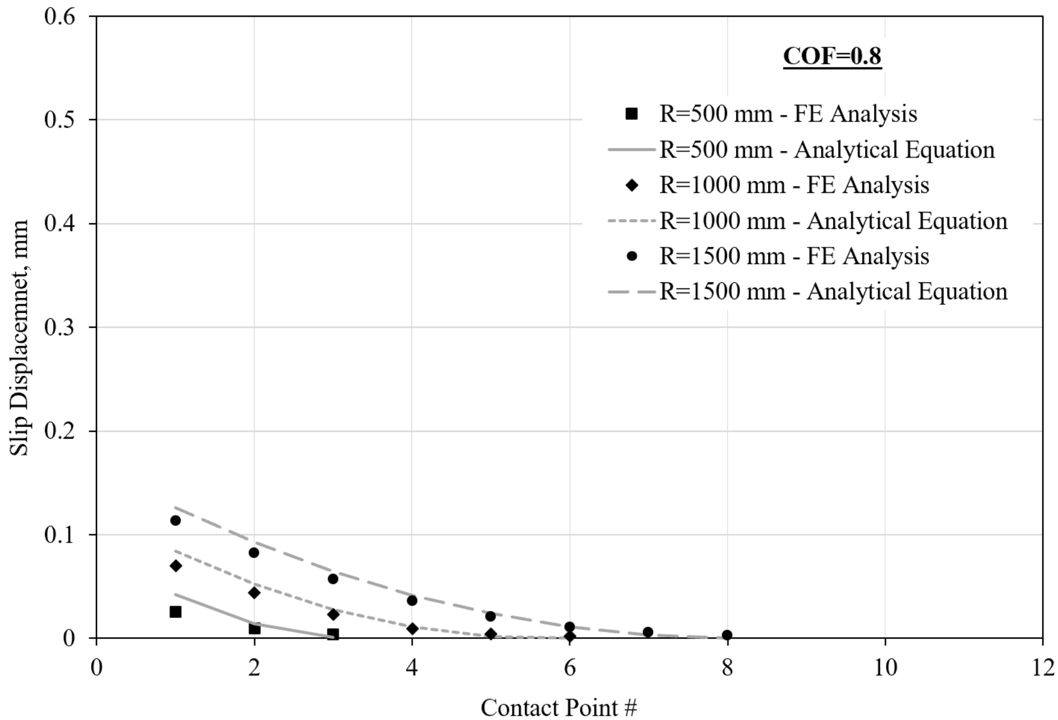


Figure 3-8 Slip displacement results for a COF of 0.8.

3.3.3 Contact force evaluation

The closed-form equation developed for the contact force is as follows (Mohareb et al. 2017):

$$F = \frac{S \cdot l}{R \cdot n} \qquad \text{3-4}$$

where S is the cable axial load, l is the lay length of the wire, R is the saddle radius, and n is the number of outer wires. l and n for the studied cable are 216 mm and 6 respectively. During the loading, S at the central angel θ can be determined based on the loading and the COF as follows:

$$S_L(\theta) = S_{max} \cdot e^{-\mu \cdot \theta} \qquad \text{3-5}$$

where S_{max} is the maximum axial load, and μ is the COF.

The results of these equations and the FE model for three different saddle radii of 500, 1000, and 1500 mm and COFs of 0.2, 0.4, 0.6, and 0.8 are shown in Figure 3-9 to Figure 3-12. It should be noted that only the point in the active region, where slip occurs, and stress range exists are shown in these figures. The actual number of contact points is a function of saddle radii and the lay angle of the cables and is not a function of COF. First, it can be seen that the contact force constantly decreases along the saddle in the uniform region as the axial force of the cable decreases (see Equations 3-4 and 3-5). Comparing the results for different saddle radii shows the contact force increases with a decrease in saddle radius, looking at Equation 3-4, the contact force is seen to be proportionate to the inverse of the saddle radius. Comparing the results of the FE model with the closed-form equation shows the results in the nonuniform region, where the cable first meets the saddle, are fluctuating around the results of the analytical equations. However, in the uniform region, the results are very close. The second contact force was found to be critical based on the FE analysis. In the nonuniform region, at the first contact points, the difference between the contact forces from the FE model and the analytical solution was lower than 12% in all cases. The maximum difference was 7%, 9%, and 12% for saddle radii of 500, 1000, and 1500 mm respectively. In the uniform region, the difference was lower than 1% in all cases.

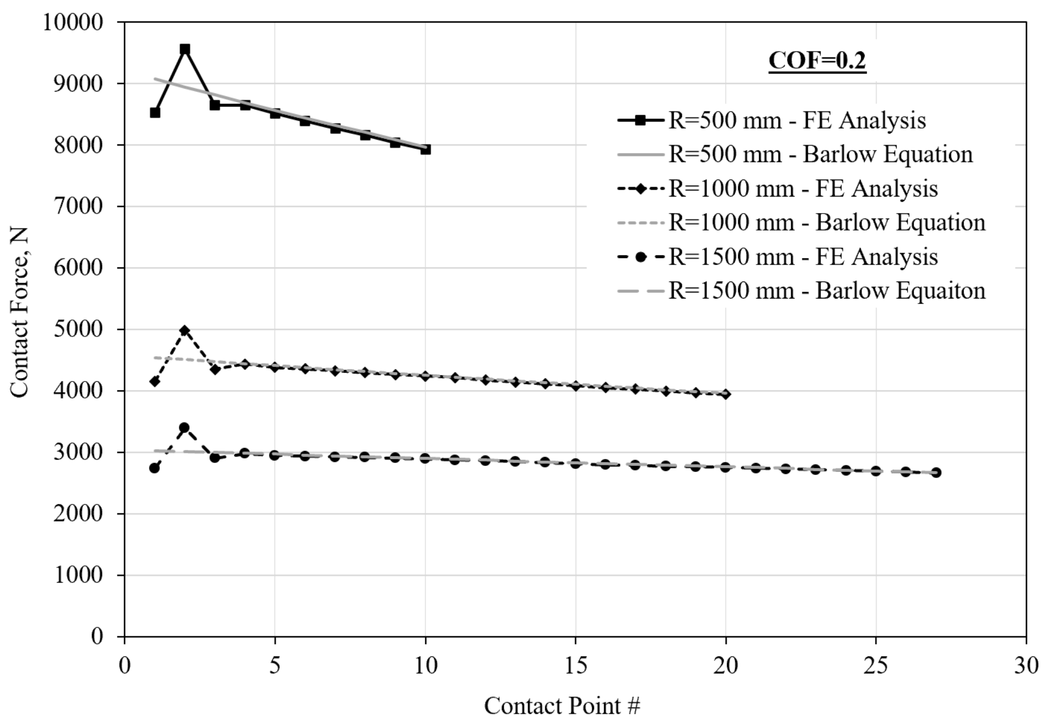


Figure 3-9 Contact force distribution for a COF of 0.2.

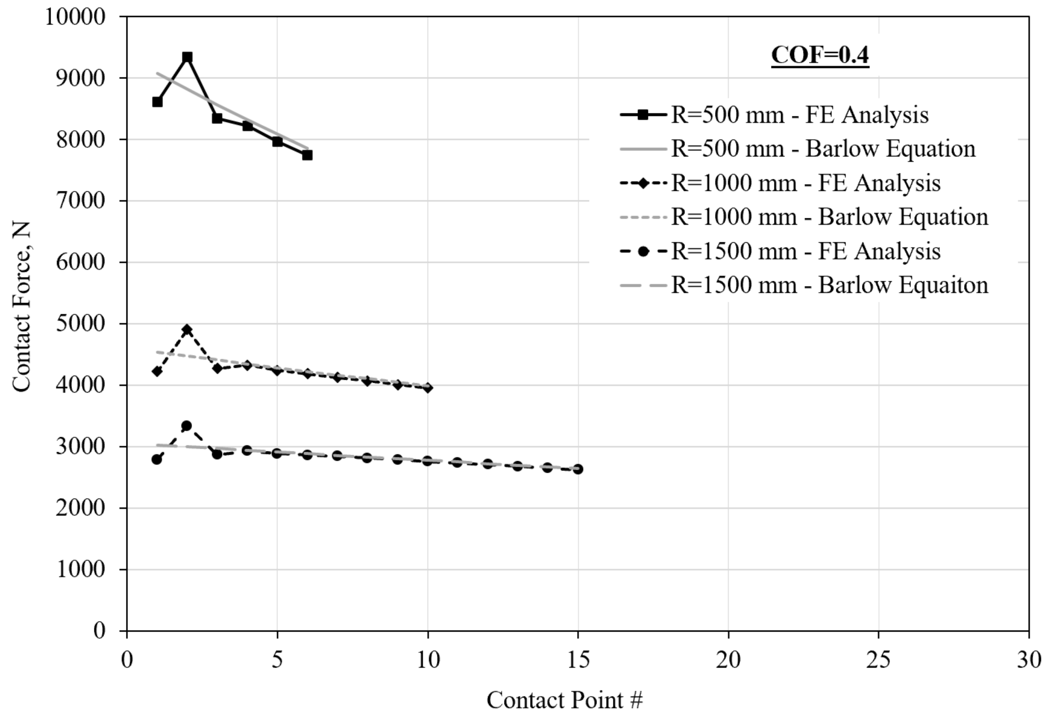


Figure 3-10 Contact force distribution for a COF of 0.4.

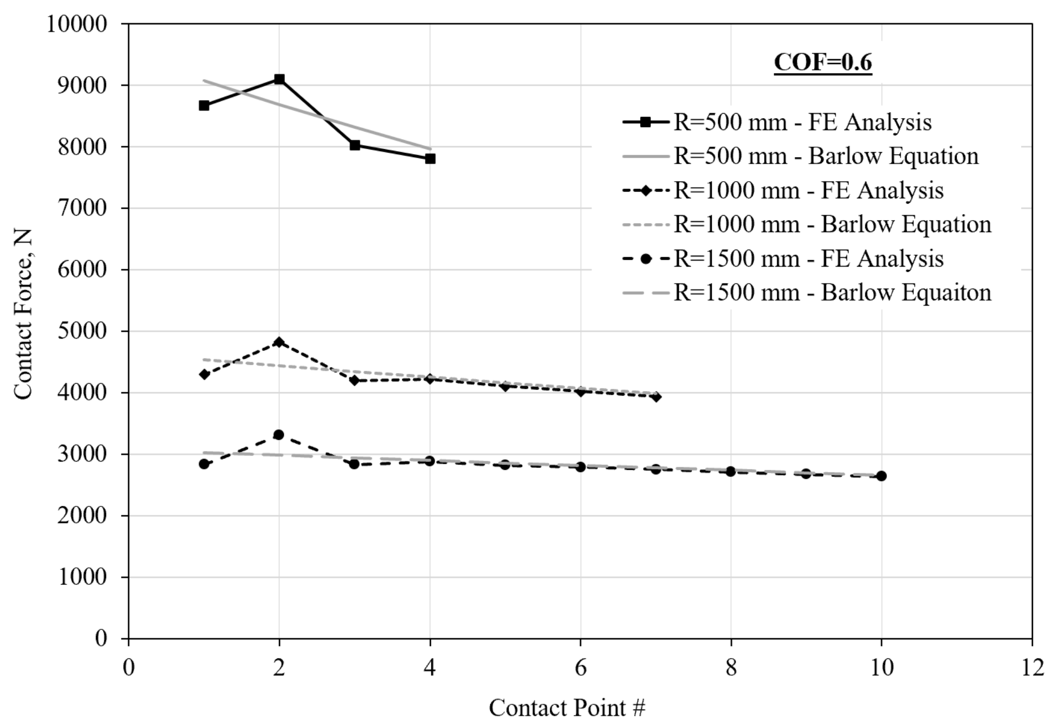


Figure 3-11 Contact force distribution for a COF of 0.6.

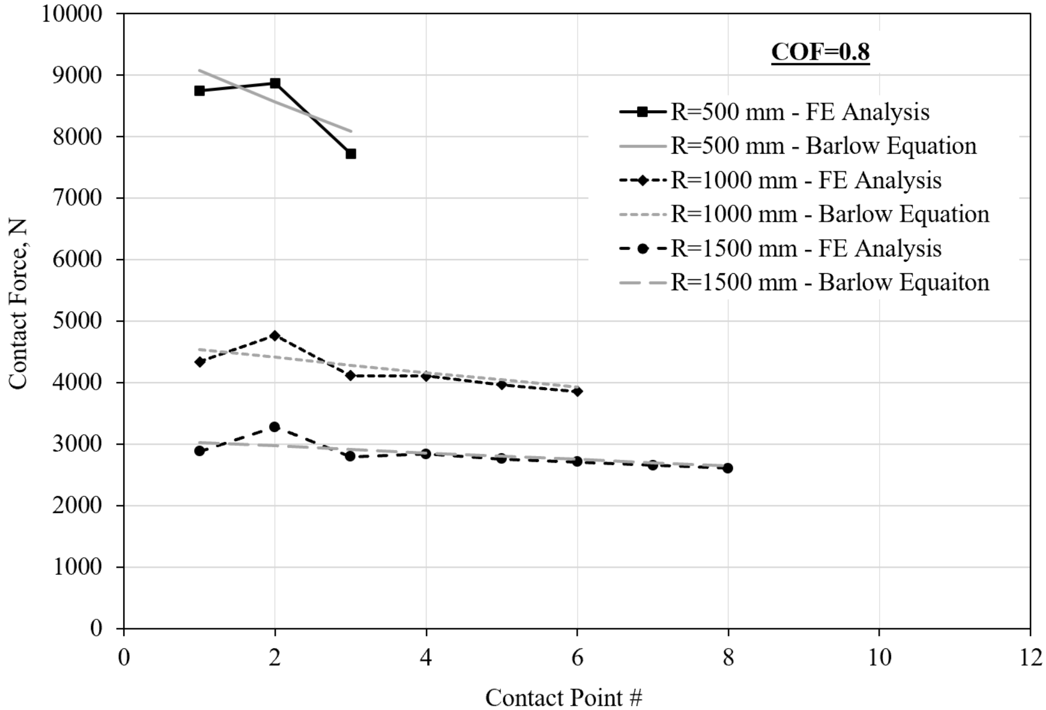


Figure 3-12 Contact force distribution for a COF of 0.8.

3.4 Fatigue parameters estimation

The Coffin-Manson parameters required for fatigue life determination can be determined using two methods: first, based on fatigue results of fully reversed tests (R = -1) with smooth specimens; second, using empirical equations in the literature, which are based on ultimate tensile strength, hardness or other more easily obtained properties of the material.

3.4.1 Estimating Coffin-Manson parameters based on fatigue tests

Coffin-Manson parameters required for fatigue life determination can be determined as follows (Roessle and Khosrovaneh 1999, Yu et al. 1991, Dowling 1998): The fatigue strength coefficient (σ_f') and fatigue strength exponent (b) are the intercept and slope of the best-fit line of the stress amplitude $\left(\frac{\Delta\sigma}{2}\right)$ versus the number of reversals ($2 \cdot N_f$) data when plotted in a log-log scale:

$$\frac{\Delta\sigma}{2} = \sigma_f' \cdot \left(2 \cdot N_f\right)^b \tag{3-6}$$

Dividing both sides by elastic modulus, E, results in an elastic strain range $\left(\frac{\Delta\varepsilon_e}{2}\right)$ versus the number of reversals ($2 \cdot N_f$) relationship:

$$\frac{\Delta\varepsilon_e}{2} = \frac{\sigma_f'}{E} \cdot \left(2 \cdot N_f\right)^b \tag{3-7}$$

The fatigue ductility coefficient (ε_f') and fatigue ductility exponent (c) are the intercept and slope of the best-fit line of the plastic strain amplitude $\left(\frac{\Delta\varepsilon_p}{2}\right)$ versus the number of reversals ($2 \cdot N_f$) data when plotted in log-log scale:

$$\frac{\Delta\varepsilon_p}{2} = \varepsilon_f' \cdot \left(2 \cdot N_f\right)^c \tag{3-8}$$

3.4.2 Closed-form equations in the literature

If the stress-life data of a particular material is not available, then the Coffin-Manson constants may be estimated using one of the several empirical equations available in the literature relating these parameters to more easily obtained information such as the material ultimate tensile strength (σ_u), hardness (HB), and fracture strain (ε_f) (see Table 3-1).

Table 3-1 Empirical equations for Coffin-Manson parameters of steel.

Method/ Parameter	$\sigma' f (MPa)$	b	ε'_f	c
(Manson 1965)	$1.9 \cdot \sigma_u$	-0.12	$0.76 \cdot \left[\mathrm{LN}\left(\frac{1}{1-RA}\right)\right]^{0.6}$	-0.6
(Mitchell et al. 1977)	$\sigma_U + 345$	$\frac{1}{6} \cdot \mathrm{LOG}\left(\frac{0.5 \cdot \sigma_U}{\sigma_U + 345}\right)$	ε_f	-0.6 for ductile and -0.5 for strong steel
(Rossele and Fatemi 2000)	$4.25 \cdot HB + 225$	-0.09	$[0.32 \cdot HB^2 - 487 \cdot HB + 191000] / E$	-0.56
(Meggiolaro and Castro 2002)	$1.5 \cdot \sigma_U$	-0.09	0.45	-0.59

3.4.3 Estimated Coffin-Manson parameters

In this study, fatigue tests have been done to determine the Coffin-Manson parameters. The tests have been done using two bridge stay cable types, one without coating and another one with galvanization. However, for the fatigue tests, hourglass samples of the wires were used. Therefore, the galvanization layer was removed for these tests. More detail about these tests can be found in Chapter 5. The results of these tests are shown in Figure 3-13. The Coffin-Manson parameters were determined by fitting a straight line to elastic strain/reversals data and plastic strain/reversals data using Equations 3-7 and 3-8. The results are summarized in Table 3-2.

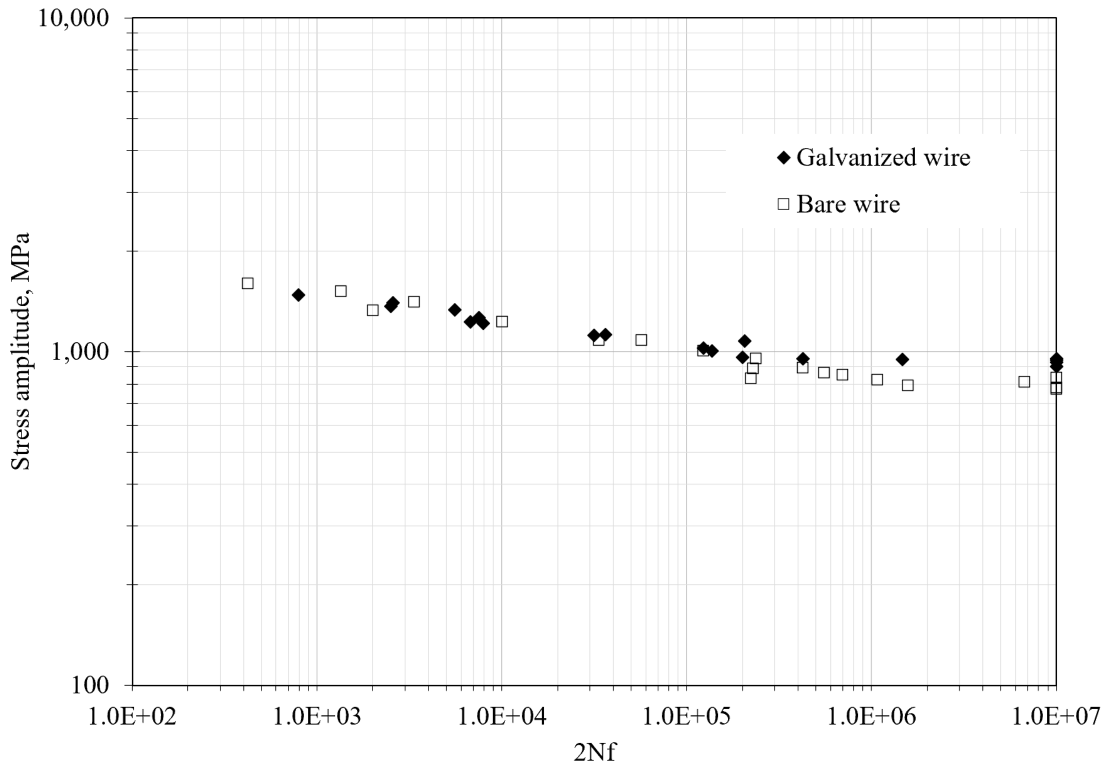


Figure 3-13 Stress vs. the number of reversals to failure data for the studied wires.

Table 3-2 Coffin-Manson parameters of the studied wires.

Specimen	Galvanized	Bare
Fatigue strength coefficient, σ'_f (MPa)	2183	2675
Fatigue strength exponent, *b*	-0.0657	-0.0859
Fatigue ductility coefficient, ε'_f	1.99	0.2067
Fatigue ductility exponent, *c*	-0.8092	-0.5047

3.5 Fatigue life estimation method

Fretting fatigue analysis employing multiaxial stress approaches are reviewed in several references (e.g. Ding et al. 2011, Araujo et al. 2002, Sum et al. 2005). (Sum et al. 2005) presented a detailed explanation of how the critical plane methods can be implemented using the SWT parameter, $\sigma_{max} \cdot \Delta\varepsilon_a$, where:

$$\mathrm{SWT} = \sigma_{\max} \cdot \Delta\varepsilon_a = \left(\sigma_f'\right)^2 \cdot \left(2 \cdot N_f\right)^{2 \cdot b} \big/ E + \sigma_f' \cdot \varepsilon_f' \cdot \left(2 \cdot N_f\right)^{b+c} \qquad 3\text{-}9$$

In this expression, σ_{max} is the maximum normal stress on the plane of interest, and $\Delta\varepsilon_a$ is the normal strain amplitude (i.e. half of the strain range) on the same plane. In the critical plane analysis, different planes are checked, and the one with the maximum value of $\sigma_{max} \cdot \Delta\varepsilon_a$ is taken as critical. For the 3-D case, the following transformations can be used:

$$\sigma = \sigma_{11} \cdot n_x^2 + \sigma_{22} \cdot n_y^2 + \sigma_{33} \cdot n_z^2 + 2 \cdot \tau_{12} \cdot n_x \cdot n_y + 2 \cdot \tau_{23} \cdot n_y \cdot n_z + 2 \cdot \tau_{13} \cdot n_x \cdot n_z \qquad 3\text{-}10$$

$$\varepsilon = \varepsilon_{11} \cdot n_x^2 + \varepsilon_{22} \cdot n_y^2 + \varepsilon_{33} \cdot n_z^2 + \gamma_{12} \cdot n_x \cdot n_y + \gamma_{23} \cdot n_y \cdot n_z + \gamma_{13} \cdot n_x \cdot n_z \qquad 3\text{-}11$$

where: $n_x = -\sin(\theta_v) \cdot \sin(\theta_h)$, $n_y = \cos(\theta_h)$, $n_z = -\sin(\theta_h) \cdot \cos(\theta_v)$

θ_h and θ_v are varied in 5° increments and $\sigma_{max} \cdot \Delta\varepsilon_a$ is calculated for each plane. Equation 3-9 is then solved for N_f. Basically, 1296 unique planes (36×36) were evaluated to find the critical plane for each point.

3.6 Finite element (FE) analysis

3.6.1 2D FE analysis of a typical fretting fatigue problem

To gain experience with the fretting fatigue contact problems, prior to applying these methods to stay cable strands draped over saddles, a first attempt was made to implement them by analyzing a typical fretting fatigue experiment of a cylindrical pad in contact with a flat specimen. Hills and Nowell (1994) presented analytical solutions for contact pressure and shear stresses at the contact surface of this problem and these solutions were used for comparison with the FE results. Figure 3-14 shows the FE model used for this evaluation.

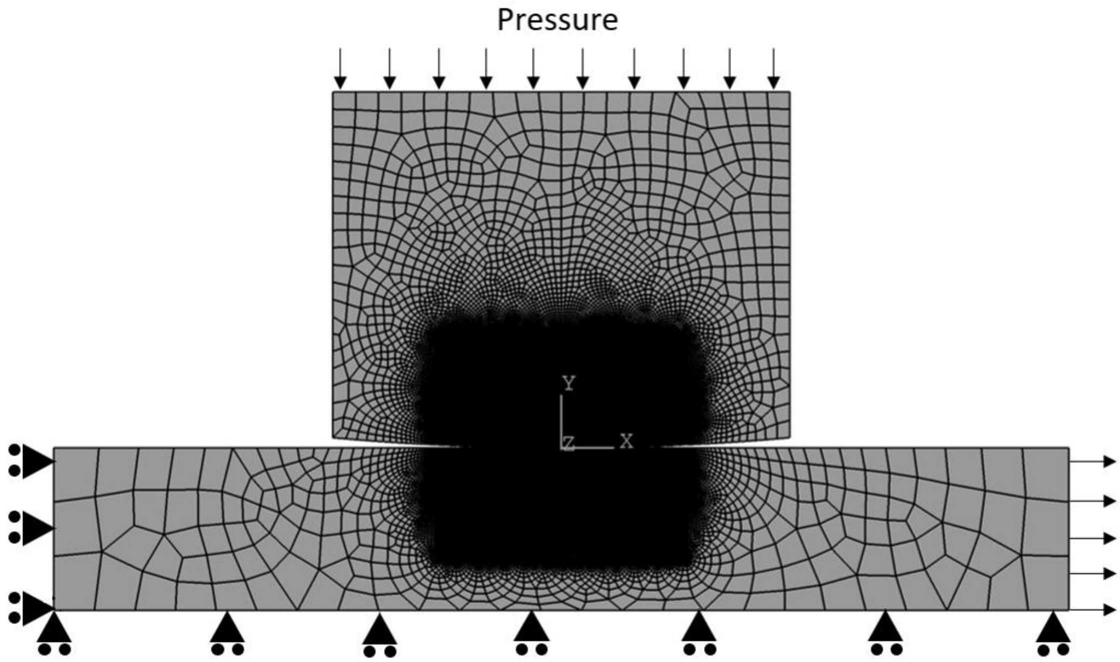


Figure 3-14 2D FE model of a cylindrical pad over a flat specimen.

The 2-D mesh used in this study consisted of four node plane strain quadrilateral (CPE4R) elements. In the contact area, regions with fine rectangular elements were defined. The element width was 10 μm, and the height varied from 10 μm at the surface to 80 μm at the inside edge of the fine mesh region. Outside of the fine mesh regions, irregular quadrilateral elements were used. Friction was imposed using the Lagrangian multiplier option, with a friction coefficient of 0.8 and default normal behaviour. An elastic material model with an elastic modulus of 126 GPa and a Poisson's ratio of 0.34 was used. The applied contact force and normal stress were 208 N and 50 MPa respectively.

The values of the maximum contact pressure from the FE analysis, 307 MPa, and the Hertzian contact pressure, 305 MPa, are close to each other. The shear stress distribution results from the FE analysis and analytical solution are compared in Figure 3-15. In this figure, a is the contact zone semi-width, x is the distance from the centre of this zone, q is the shear stress at the surface, and f_p is the peak contact pressure.

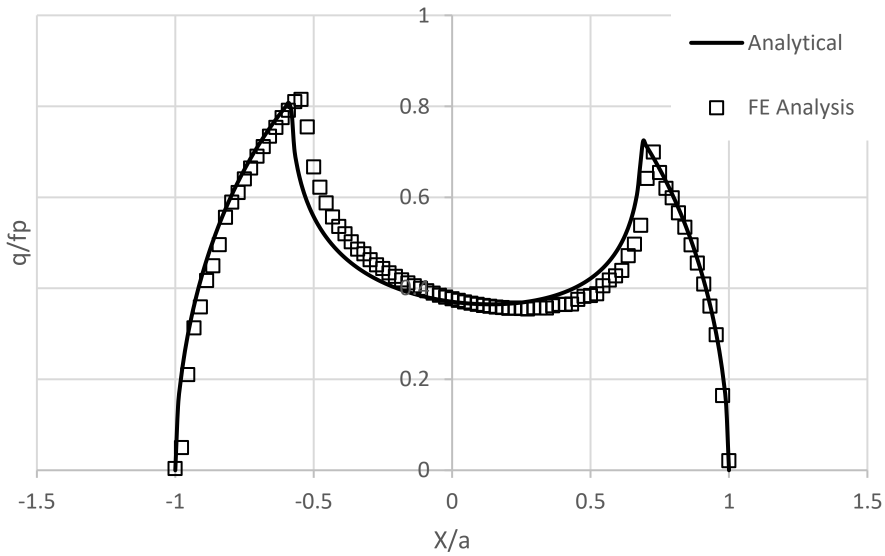


Figure 3-15 Shear stress based on the analytical solution in Hills and Nowell (1994) versus FE model result.

3.6.2 3D FE model of the contact point between a cable and a saddle

An interface model for a contact point between a cable and a saddle was first presented in Mohareb et al. (2017). Elastic material properties were used in this model for both wire and saddle. Chehrazi et al. (2020) employed a similar model with plastic material properties for the saddle to consider the effect of plastic deformation on the saddle part. The model employed in Chehrazi et al. (2020) was used in the current thesis. Details of this model were as follows: the ABAQUS FE program was used to formulate this problem. Contact force, slip displacement, and normal stress are the inputs of this model.

The FE model and the boundary conditions are shown in Figure 3-16 (a). To reduce the analysis time, a plane of symmetry was assumed, and to reduce the height of the saddle, a 0.5 mm thick stiffened layer was added at the bottom of the saddle part. This stiffened region prevents excessive bending of the saddle as well. The wire part is curved to model a discrete contact point, the curvature of the wire part is equal to the curvature of an outer wire in a cable on a straight surface. The curvature on the saddle part was assumed to be high enough to be ignored in the modelling. Tangential contact surface behaviour was controlled by the penalty method. Assuming hard contact in the interface, the normal contact method was imposed using the penalty method. Eight node linear brick elements (C3D8R) with an approximate size of 25 μm ×25 μm were used in the contact surfaces of both parts. For the high-strength steel cable, a linear material model of steel with an elastic modulus of 200 GPa and a Poisson ratio of 0.3 was assumed. A nonlinear material model, elastic-perfectly plastic, of S235 steel with an elastic modulus of 200 GPa, a Poisson's ratio of 0.3 and a yield stress of 500 MPa was assumed for the saddle material.

The loading was applied in three steps. First, a small amount of contact force was applied to ensure the contact (specifically, a contact pressure at the bottom of the saddle part equal to 1 MPa). Then, the maximum contact force, axial stress range, and slip displacement associated with the loading phase are applied. The axial load was applied to the end of the wire and the slip displacement was applied by moving S1 and S3 surfaces. In the last step, the critical parameters reduce to their values for the unloading phase. In this step, S1 and S3 surfaces return to their original position. These loading steps are shown in Figure 3-16 (b). Also, it was seen that restrains on W4 and S4 surfaces do not significantly affect the analysis and change the results. The W3 surface was fixed in x' and y' directions and it only can be moved in the z' direction at the top. The initial increment size in all steps was set to 0.001. The maximum step size was set to 0.1. the number of increments varies based on the convergence in increments. The direct equation solver with the full newton solution technique implemented in ABAQUS was used for the analysis.

The loading and unloading stages were applied in two different steps to enable the recording of the stress/strains at the end of each stage. The results at the end of the second (loading) and third (unloading) steps were used to determine the SWT parameter. Figure 3-17 shows sample results for the SWT parameter along the wire. The maximum value of the SWT parameter was used to determine the fatigue life based on Equation 3-9.

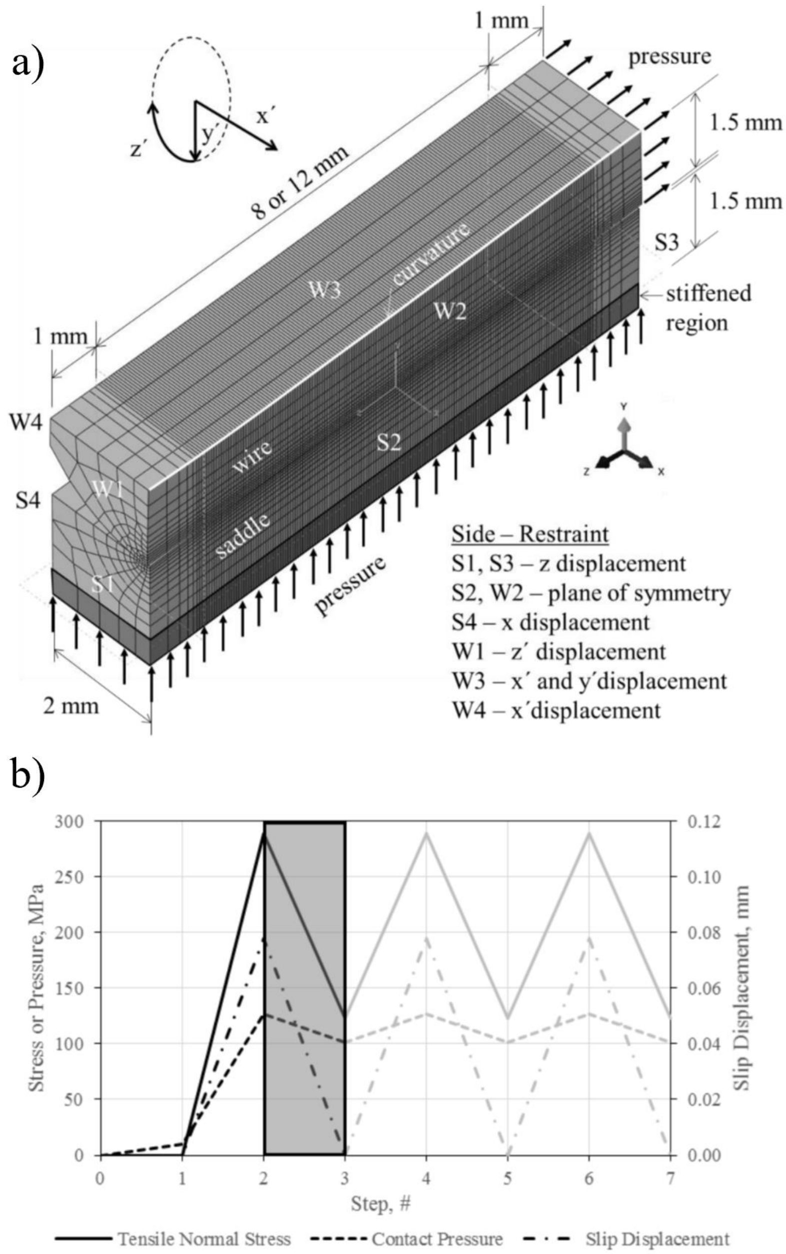


Figure 3-16 3D finite element model of the contact point (a), and loading steps (b) (Walbridge et al. 2017).

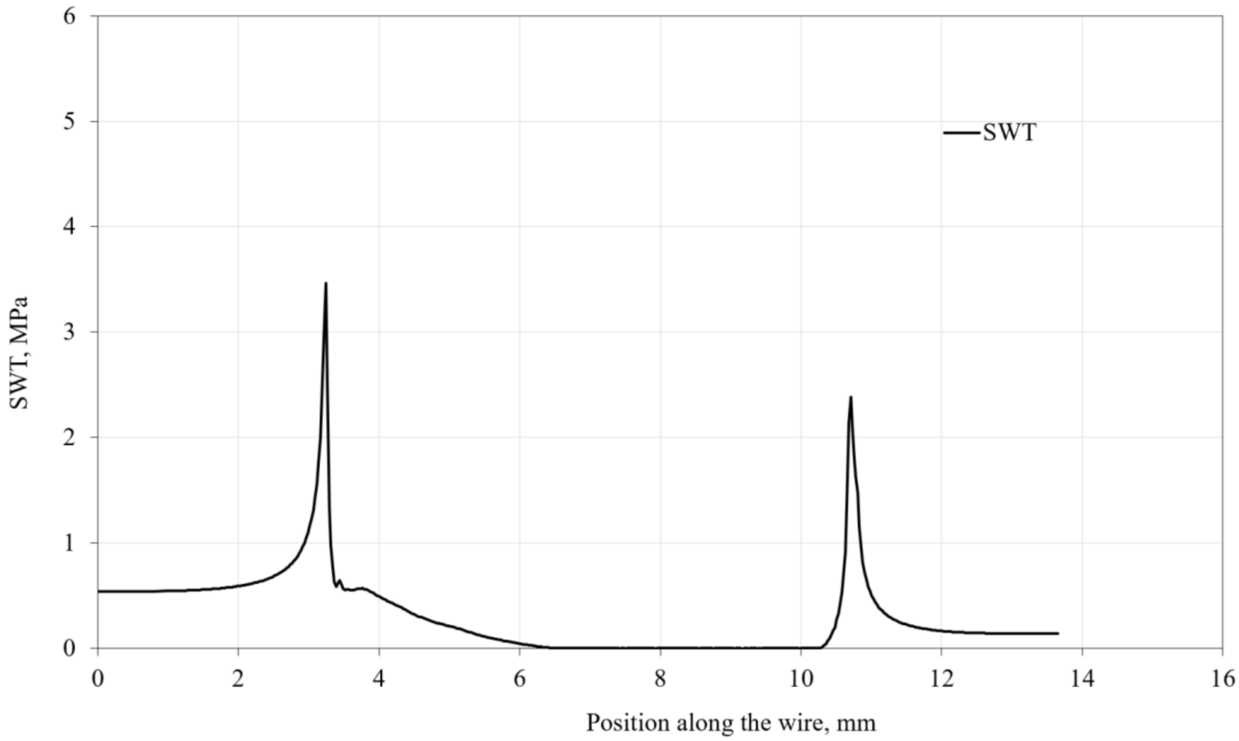


Figure 3-17 SWT parameter along the wire at the contact point for the first point of the saddle with R = 1000 mm.

3.7 Fatigue life prediction

The discussed framework was used to evaluate the fretting fatigue life of the cables in the tests at TU Berlin for different saddle radii. The analysis was done based on a COF equal to 0.7, which was an upper limit in the tests at TU Berlin (Mohareb 2020). The analysis was done based on the closed-form equations for evaluating the critical parameters. Also, Coffin-Manson parameters from the two-wire types tested at the University of Waterloo were employed. The fretting fatigue life predictions made using the 3D FE model are summarized in Figure 3-18. Looking at this figure, general trends are predicted reasonably using the suggested modelling framework. It can be seen that lower fatigue life for the saddle radius of (R = 1500 mm) is predicted. One possible explanation for lower prediction at this saddle radius is not considering the beneficial effect of wear. Wear was not modelled in the current study. An iterative process can be used to remove the elements at the contact surface and model the wear over time. Modelling the wear can decrease the stress concentration and consequently the SWT parameter. Also, wear can remove the small cracks and increase the fatigue life for the tests in air (Vingsbo and Soderberg 1988). The amount of slip displacement and wear is higher for the tests at R = 1500. Apart from modelling the wear, it is believed that further refinements could lead to improved predictions, such as: using material properties based on tests of the actual wire material used in the fatigue tests at TU Berlin, evaluating the possible defect of the wires, and considering the possible effects of the thin galvanizing layer on the outer surface of the wire.

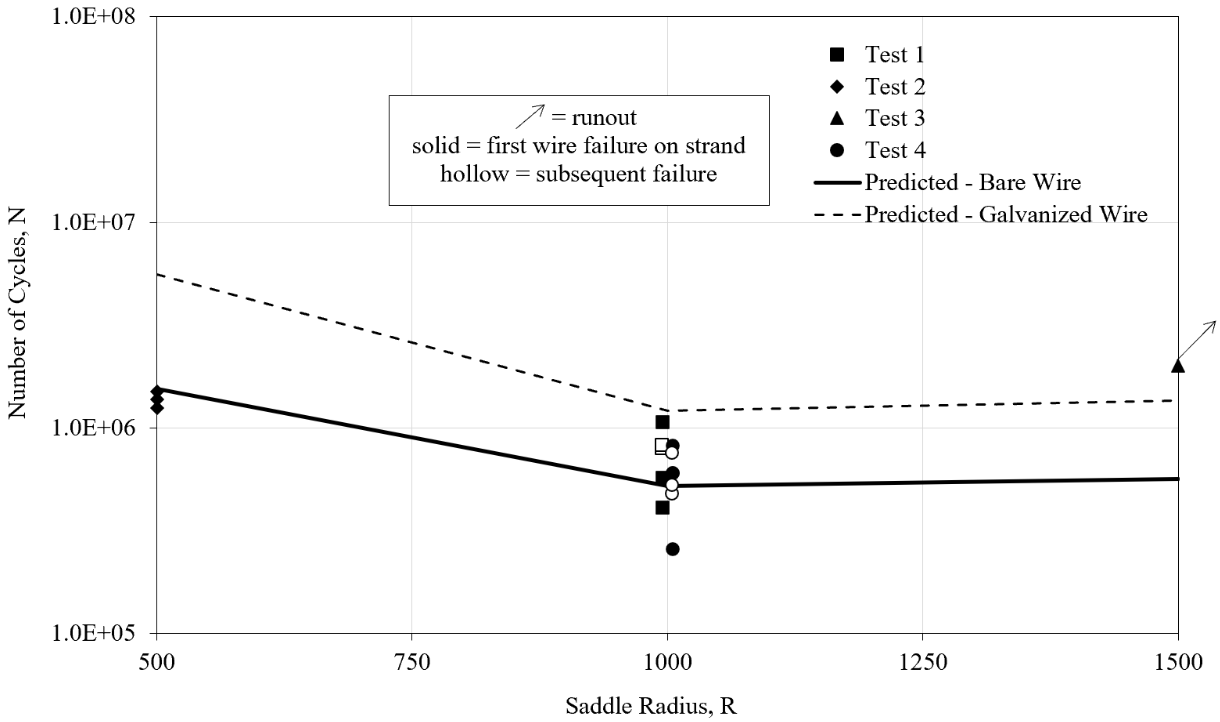


Figure 3-18 Pilot fatigue test results versus predictions based on the analytical equations for determining the critical parameters.

3.8 Fretting maps

The current analysis procedure is complicated and very time-consuming. Each analysis takes around 6 to 12 hours. Therefore, it would not be practical for routine design purposes, and a simpler approach is required. To investigate the feasibility of using the presented framework to develop generalized design tools, several analyses were performed to generate fretting maps, which can be used for design. In fretting maps, the fatigue life or a damage parameter, e.g. SWT, is plotted against other critical parameters, such as the slip displacement, the normal stress or the contact force. Interpolation can be used with these maps to find the fatigue life for a set of new parameters.

To generate fretting maps in the current work for the fretting fatigue problem of a cable over a saddle, analyses were done for a range of contact forces and slip displacements for the first point of tests at TU Berlin with the saddle radius of 1000 mm. The normal stress was the same for all analyses (100-300 MPa). Given the uncertainties in the COF, the maps were generated for four different COFs of 0.2, 0.4, 0.6, and 0.8. SWT results are plotted for a range of slip displacements and contact forces in Figure 3-19. With these maps, the fatigue design only requires the evaluation of the critical parameters (e.g., contact force, slip displacement), which is much simpler for design purposes. With the critical parameters, the SWT parameter can be determined from the fretting maps. Then, the fatigue life can be determined using Equation 3-9, thus avoiding the need for a new FE analysis.

These maps show a possible simplified tool for design purposes. However, these maps are only applicable to a single stress range (200 MPa) and mean stress. While the stress range is typically 200 MPa for the design of saddle systems, the mean stress changes for each saddle radius. Therefore, more maps that cover a wide range of mean stress are required for a generalized design tool. Effects of coating and material properties are other parameters that require further investigation in order to make the fretting maps useful as a generic design tool.

Looking at the fretting maps, it is seen that the SWT parameter increases with an increase in COF, slip displacement, and contact force. However, it should be noted that these parameters depend on each other and typically a higher COF is associated with a lower slip displacement.

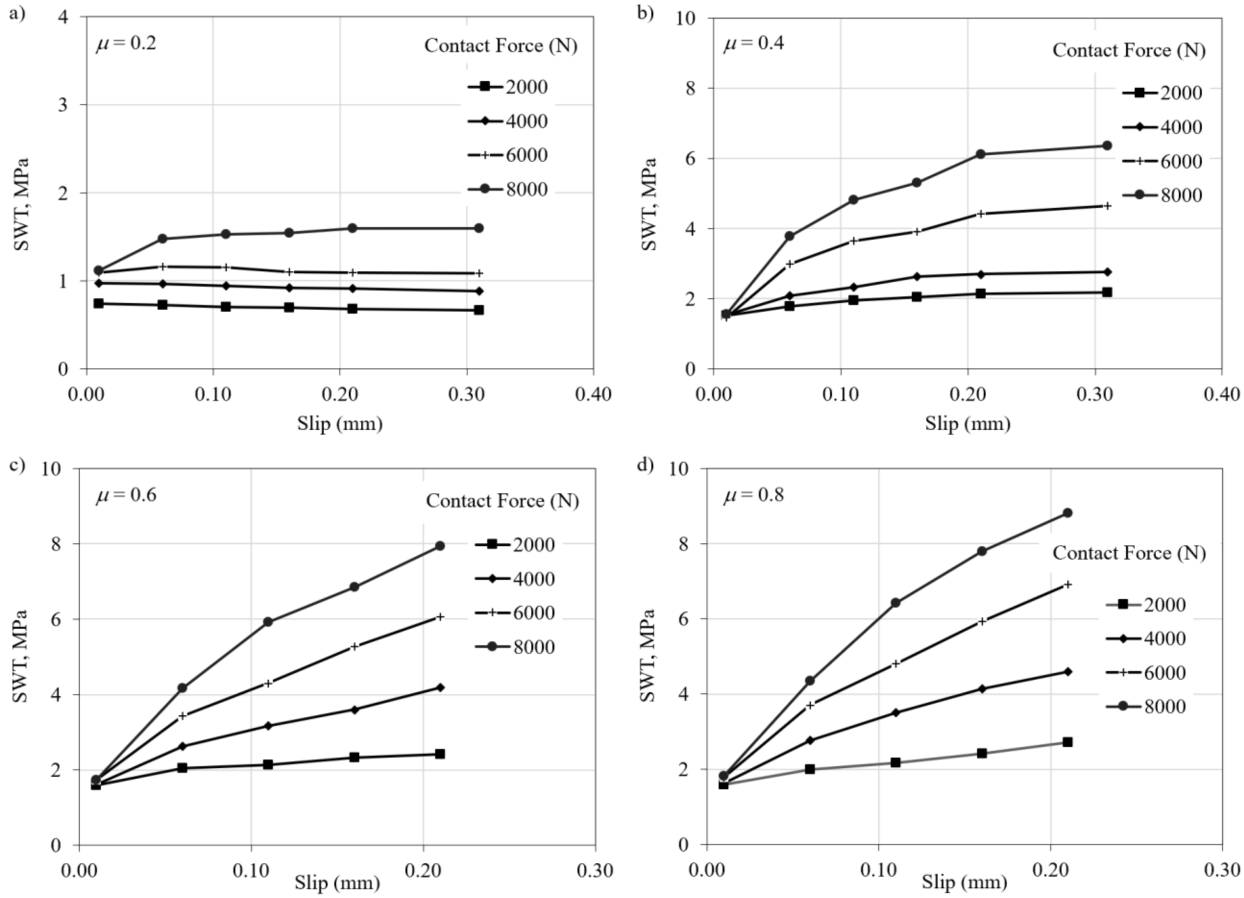


Figure 3-19 Fretting maps.

3.9 Wear modelling

Large-scale fretting fatigue tests at TU Berlin showed that the volume of wear is typically higher at the first contact points. It was argued that wear at these points can affect the contact forces or slip displacements. Therefore, it was decided to use the 2D model to evaluate the possible effect of wear on these parameters and consequently the fretting fatigue life of the cables.

Archard's equation has been used in different engineering fields to determine the wear volume in the contact area between two contacting components. In this paper, the numerical approach of Archard's equation developed by McColl et al. (2004) was used to model the wear at the contact points of saddle systems. Based on Archard's equation, the wear volume can be determined using the following equation (Archand 1953):

$$\frac{V}{S} = K \cdot \frac{P}{H} \qquad 3\text{-}12$$

where V is the wear volume, S is the sliding distance, K is the wear coefficient, P is the contact force, and H is the hardness of the material. For a local point at the contact region, Archard's law can be expressed by the following equation (McColl et al. 2004):

$$\frac{dh}{ds} = k_1 \cdot p(x) \qquad 3\text{-}13$$

where k_1 is the local wear coefficient, and $p(x)$ is the local contact pressure. (McColl et al. (2004) presented a numerical method for determining the wear depth at a point in the contact surface between two components. At a given point, the increment of the wear depth, Δh_i, can be determined using the following equation:

$$\Delta h_i = \Delta N \cdot k_1 \cdot p_i \cdot \delta_i \qquad 3\text{-}14$$

where p_i and δ_i are the pressure and the slip displacement per cycle at Point i, and ΔN is the increment in the number of cycles.

In this section, the 2D discrete FE model (shown in Figure 3-4) was used to evaluate the effect of wear on the contact force distribution between the contact points. The wear at each contact point was modelled by displacing the center of each arch in the 2D FE model. Equation 3-14 was used for wear modelling. The exact evaluation of the local wear coefficient, k_1 requires wear tests that have not been done in this work. However, looking at Equation 3-14, $\Delta N \cdot k_1$ is similar for all the contact points. Thus, the contact pressure and the slip displacement are the effective parameters of this equation. These parameters can be employed to determine relative wear depths at the contact points. Therefore, instead of evaluating wear depth for each contact point independently, a relative wear depth, $p_i \cdot \delta_i$, was calculated for each point. In each iteration, the wear increment was assigned to the point with the maximum value of $p_i \cdot \delta_i$ and the wear depths of the other points were calculated based on their relative wear depth.

A Python script was used to automate the iterative process required for this modelling procedure. The analysis started with a model without wear. Following each analysis, the contact forces and the slip displacements were stored. Then, a new FE model was created based on the results from the previous model. Based on the slip displacements and the contact pressures at the contact points, the rigid arches on the saddle part were displaced to model the wear. For each contact point, the incremental wear depth was based on the relative wear and the wear increment for each iteration. Using this approach, the distribution of the contact forces can be determined based on the maximum final wear depth at the contact points (see Figure 3-20). For this approach, a wear increment of 0.02 mm was used and the analysis was performed until maximum wear of 0.4 mm at the first contact point was achieved.

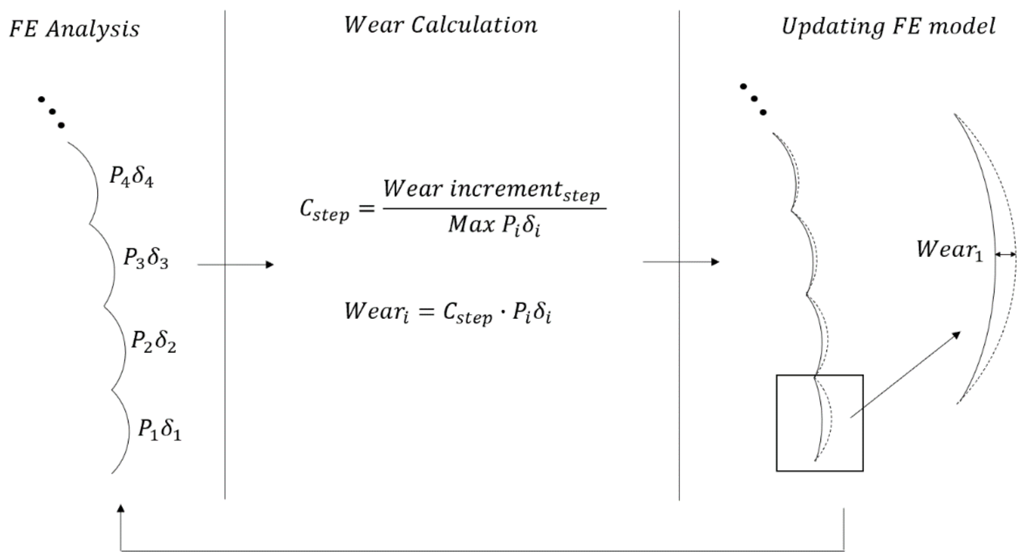


Figure 3-20 Wear modelling procedure.

Evaluating the contact pressure and slip displacement at each contact point is the main task in the wear modelling process. The slip displacement results from the 2D FE model were used. Although the 2D FE model results can be used for calculating the contact forces, determining the contact pressure at a contact point requires a more detailed FE model. So, the interface model shown in Figure 3-16 was employed to determine the contact pressure for each contact force.

Despite the higher accuracy of the 3D model, running the 3D model multiple times after each analysis of the 2D model is time-consuming. To overcome this issue, the contact pressure at the center of the contact point was determined for a few contact forces, and interpolation was used to find the values in between. Figure 3-21 shows the contact force vs. contact pressure results. Note that this is not a simple relationship due to the complex geometry (doubly-curved contact surface) and nonlinearity (plastic deformation of saddle material) associated with this problem.

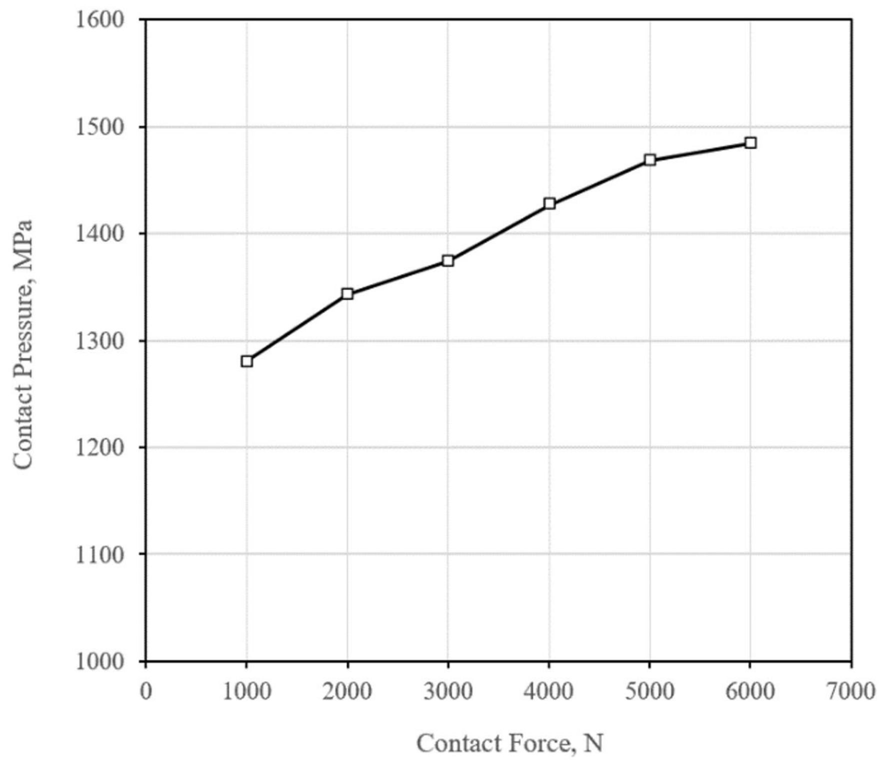


Figure 3-21 Contact force versus contact pressure.

Figure 3-22 and Figure 3-23 show the effect of wear on contact forces and the slip displacements for a saddle radius of 1000 mm and a COF of 0.7. (Results for COFs of 0.2, 0.4, 0.6, 0.8 and saddle radii of 1000 and 1500 mm are shown in Appendix A.) As can be seen, the effect of wear on slip displacement is not significant. However, the contact force results in the nonuniform region can considerably change by modelling the wear. The contact force at the first point decreases, while the contact force at the other points increases, especially at the second contact point.

Contact force is a critical parameter in determining the fretting fatigue failure. The fretting fatigue test results at TU Berlin show that the cable usually fails at the second contact point. The contact force results presented in Figure 3-22 offer a possible explanation for the failure of the cables at the second contact point. The results of critical parameters were used with the 3D FE model of the contact point to determine the SWT parameter and the fretting fatigue life. SWT and fatigue life results based on each method are summarized in Figure 3-24 and Figure 3-25 for the saddle radii of 1000 mm and the COF of 0.7. As can be seen in these figures, the critical point is the second point when the wear is modelled.

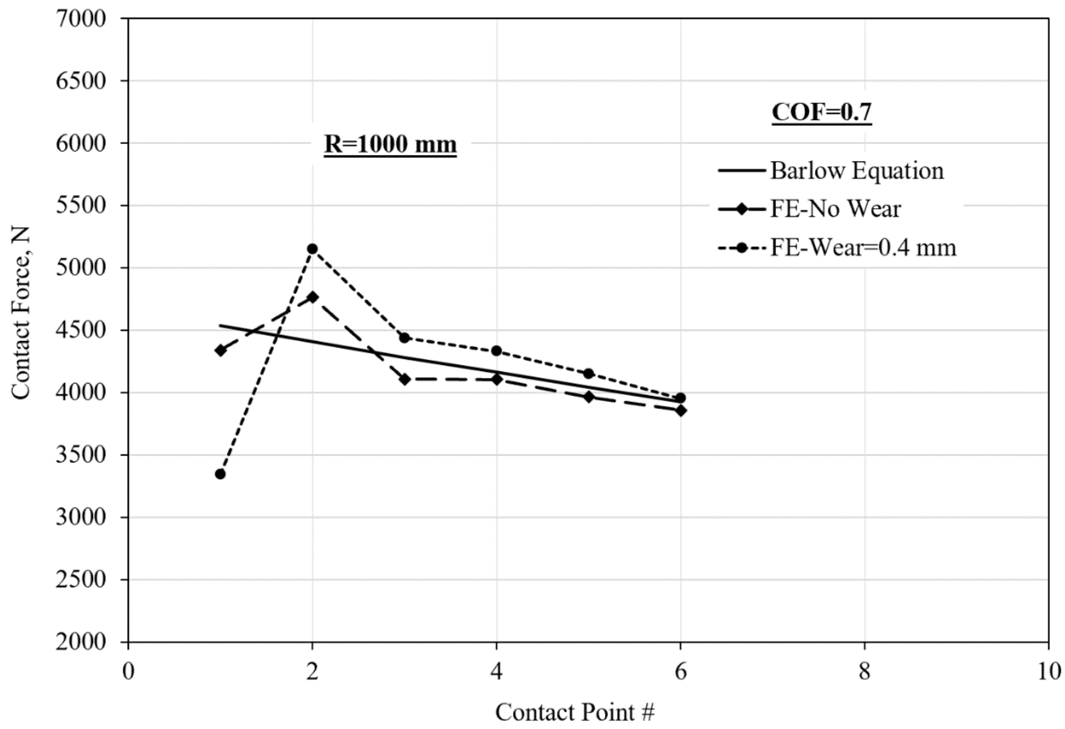


Figure 3-22 Contact force results for R=1000 mm and COF=0.7.

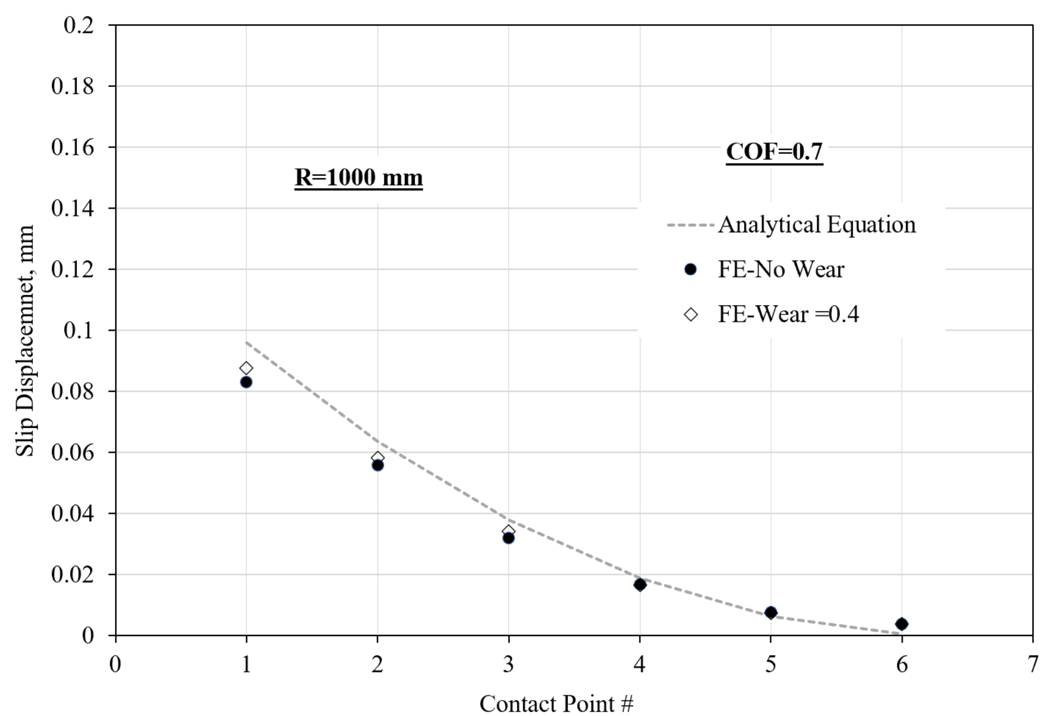


Figure 3-23 Slip displacement results for R=1000 mm and COF=0.7.

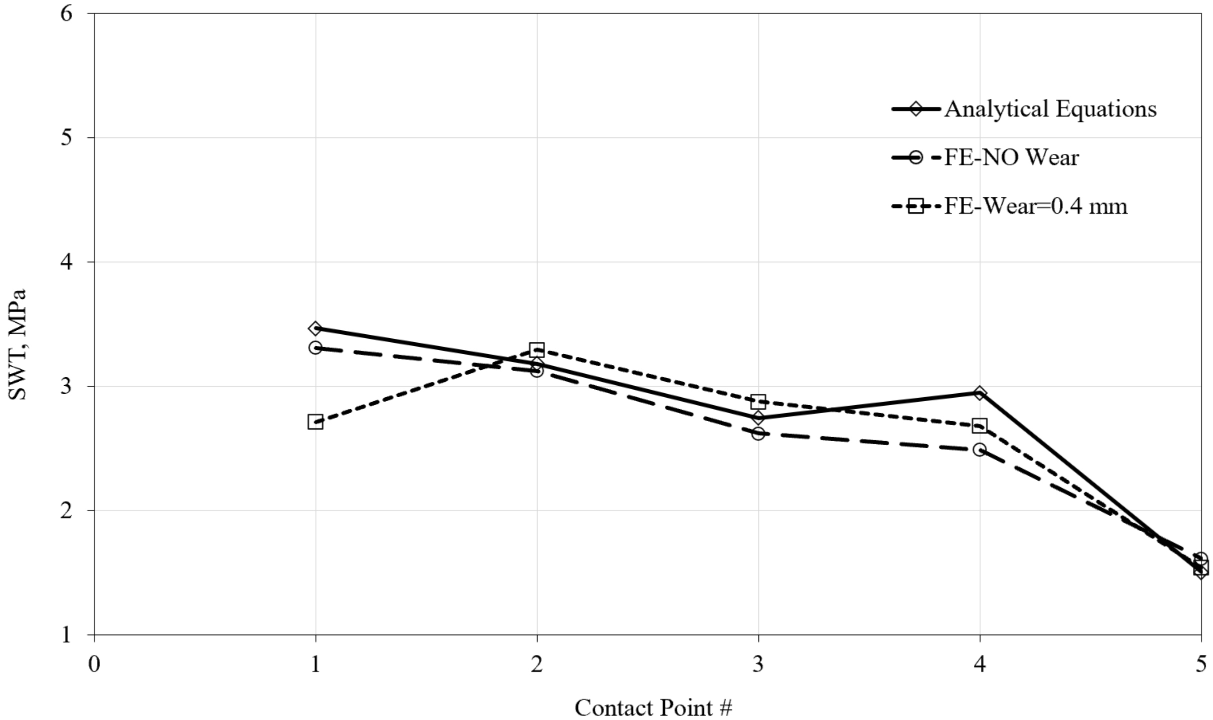


Figure 3-24 SWT results for R=1000 mm and COF=0.7.

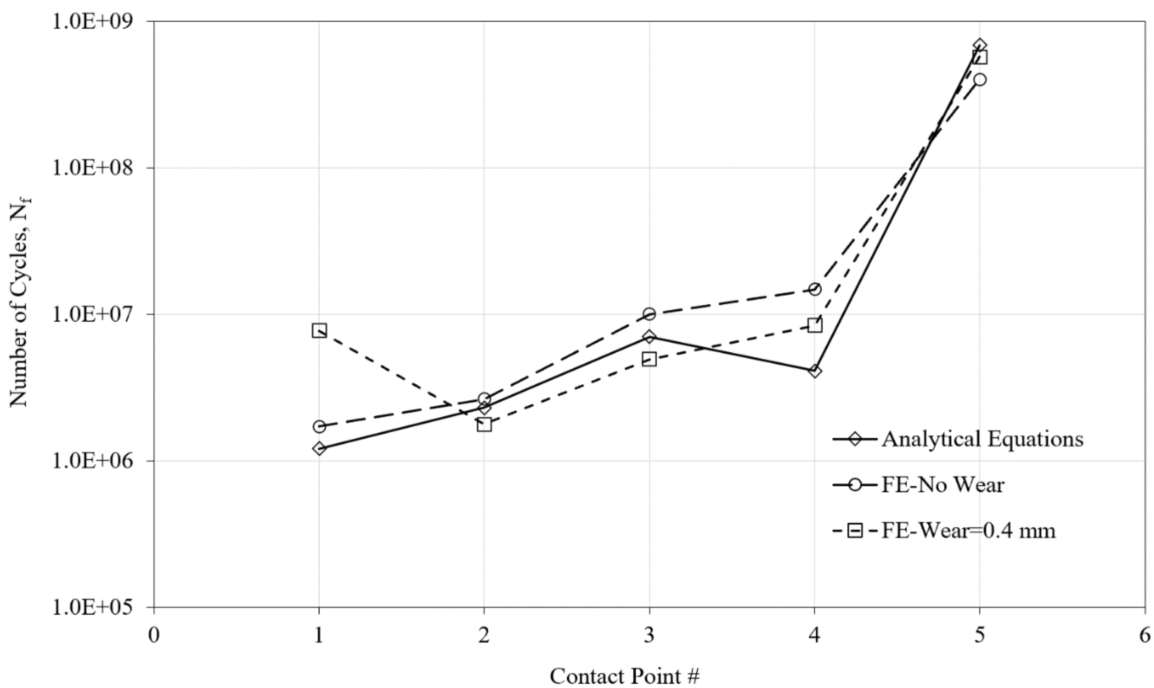


Figure 3-25 Fretting fatigue life results for R=1000 mm and COF=0.7.

4. Probabilistic Fretting Fatigue Analysis of Bridge Stay Cables at Saddle Supports

This chapter starts by presenting methods used in the current study for probabilistic fatigue analysis of bridge stay cables at saddle supports, namely: the Multiplicative Dimensional Reduction Method (M-DRM), and Monte Carlo simulation (MCS). The problem is then defined, and the model parameters and their statistics are discussed. Following this, different approaches for probabilistic analysis of the problem are discussed. Finally, the results are presented, and a sensitivity analysis is performed to determine the critical model parameters.

4.1 Methods

4.1.1 Monte Carlo simulation

Details of MCS can be found in a number of references (e.g., Ang and Tang 1984, Melchers 1999). MCS is a numerical method that can be used to find the probability of failure, a histogram of a response, or the cumulative density function (CDF) of a response by evaluating the response function of interest numerous times. In each trial, a random vector is generated for the input variables. The response function is then evaluated with trial values for the inputs, and the output/result is stored. Finally, the stored results can be used to determine the probability of failure or other parameters of interest. A conceptual explanation of MCS is shown in Figure 4-1. In this figure, F(*S*) and F(*R*) are CDFs of load (solicitation) and resistance in a simple structural problem, and *u* is random number uniformly distributed between zero and one, used to generate trial values of *S* and *R* (solicitation and resistance) for each MCS "trial".

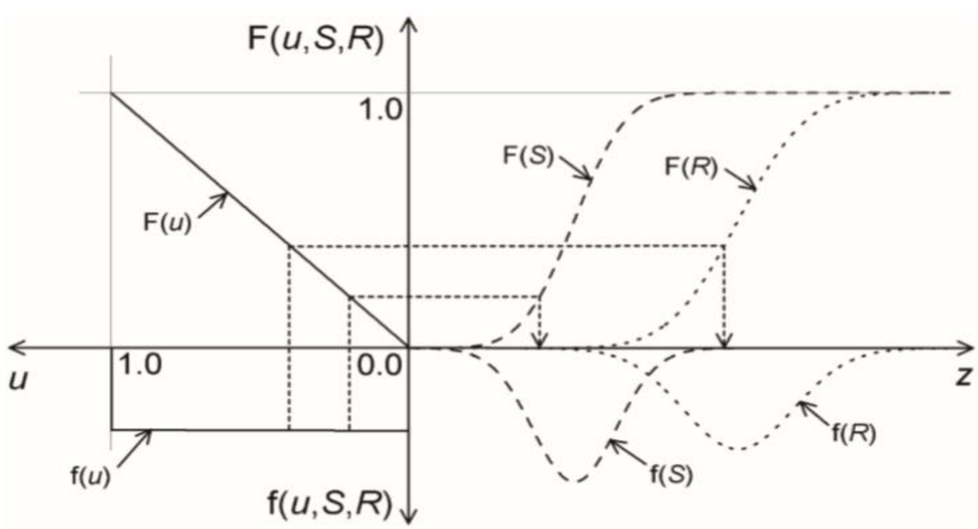


Figure 4-1 Conceptual explanation of MCS (Walbridge 2005).

The accuracy of the results from MCS increases with the number of trials. More trials are required when MCS is used to determine the probabilities of failure associated with rare events. For example, 200 trials can be reasonable when the objective is evaluating an event with a probability of 0.5 but it is not acceptable for evaluating an event with a probability of 0.01. In other words, with a low number of trials, the tails of the PDF/CDF or histogram of the output are not reliable. The required number of trials can be calculated as follows:

$$\mathrm{N} \approx \frac{1}{P_f \cdot \mathrm{COV}^2} \qquad \text{4-1}$$

where P_f is the probability of failure and *COV* is the coefficient of variation. As can be seen in Equation 4-1, the main disadvantage of MCS is the large number of trials needed for high accuracy. This can be very problematic when the analysis time for each trial is high (e.g., if an FE analysis is required in each trial).

4.1.2 Multiplicative dimensional reduction method (M-DRM)

There are many problems in mechanics where similar difficulties have been faced in performing probabilistic analysis using MCS where the complexity of the problem requires that a new FE analysis be performed for each trial. To address such problems, a variety of techniques have been developed to estimate the properties of the probabilistic distribution of interest using a much smaller number of trials (e.g., MCS with importance sampling). The multiplicative dimensional reduction method (M-DRM) is a recently developed statistical method (Zhang and Pandey 2013); this method has been implemented for several engineering problems (e.g., Balomenos et al. 2015, Raimbault et al. 2015, Balomenos and Pandey 2016). With this method, based on the analysis of the results of a limited number of trials, not only are statistical properties, such as mean and standard deviation, calculated but primary and total sensitivity analysis can also be performed (Zhang and Pandey 2014). Also, the PDF of the response can be determined. The main benefit of this method is a reduction in the number of trials required. M-DRM uses Gaussian quadratures, the type of which varies depending on the distribution type. These quadratures are based on the approximation of integrations evaluated at known Gauss points. M-DRM steps are described briefly in the following paragraphs.

In probabilistic analysis, the response function is generally a function of several input variables. The following equation shows the general form of a structural response based on input variables.

$$Y = h(\mathbf{x}) \qquad 4\text{-}2$$

in which *Y* is the structure response or analysis output (e.g., fatigue life or the SWT parameter in fatigue design), and **x** is a vector of input variables (e.g., live load, dead load, coefficient of friction). The probability of failure can be calculated with the following equation:

$$p_f = \int_{\{g(\mathbf{x}) \leq 0\}} f_{\mathbf{x}}(\mathbf{x})\, d\mathbf{x} \qquad 4\text{-}3$$

in which $g(\mathbf{x})$ is the limit state function and $f_{\mathbf{x}}(\mathbf{x})$ is the joint PDF of the input variables. As described in the previous section, calculating p_f using MCS is a time-consuming task if each trial is time-consuming (e.g., if FE analysis is required in each trial). In M-DRM, the response function is approximated using the multiplication of cut functions:

$$Y = h(\mathbf{x}) \approx h_0^{1-n} \cdot \prod_{i=1}^{n} h_i(x_i) \qquad 4\text{-}4$$

where $h_i(x_i)$, the *i*th cut function, is the response when all the input variables except the *i*th variable are fixed at their mean values:

$$h_i(x_i) = h(c_1, \dots, c_{i-1}, x_i, c_{i+1}, \dots, c_n) \qquad 4\text{-}5$$

and h_0 is the response when the input variables are fixed at their mean values:

$$h_0 = h(c_1, c_2, \dots, c_n) \qquad 4\text{-}6$$

where $c_1, c_2, \dots, c_n$ are the mean values of the random variables. Based on M-DRM, the *k*th moment of the response function can be approximated as follows:

$$E[Y^k] = E\{[h(\mathbf{x})]^k\} \approx E\left\{\left[h_0^{(1-n)} \cdot \prod_{i=1}^{n} h_i(x_i)\right]^k\right\} \qquad 4\text{-}7$$

in which $E[Y^k]$ is the *k*th statistical moment. If all the input variables are independent, the simplified version of Equation 4-7 is as follows:

$$E[Y^k] \approx h_0^{k(1-n)} \cdot \prod_{i=1}^{n} E\left[\left(h_i(x_i)\right)^k\right] \tag{4-8}$$

Based on Equation 4-7, determining the k^{th} moment of the response requires calculating the k^{th} moment of all cut functions. The k^{th} moment of a cut function can be calculated as follows:

$$E\left[\left(h_i(x_i)\right)^k\right] = \int_{X_i} [h(x_i)]^k \cdot f_i(x_i) dx_i \tag{4-9}$$

Gauss quadrature formulas can be used to simplify the numerical integration and reduce the analysis time:

$$E\left[\left(h_i(x_i)\right)^k\right] \approx \sum_{j=1}^{L} w_j \cdot \left[h_i(x_j)\right]^k \tag{4-10}$$

in which x_j and w_j are the quadrature points and the weights.

The main objective of this method is to calculate the PDF (or CDF) of the response, which can then be used to perform a probabilistic analysis. The Maximum Entropy principle using fractional moments is used to find the most unbiased probability distribution of the response. Based on (Zhang and Pandey 2013), the estimated PDF of the response function is obtained as:

$$\widehat{f_Y}(y) = exp\left(-\sum_{i=0}^{m} \lambda_i \cdot y^{\alpha_i}\right) \tag{4-11}$$

in which α_i and λ_i can be found by the following optimization:

$$\begin{cases} Find: \ \{\alpha_i\}_{i=1}^{m} \quad \{\lambda_i\}_{i=1}^{m} \\ Minimize: I(\boldsymbol{\lambda}, \boldsymbol{\alpha}) = \ln\left[\int_{y} exp\left(-\sum_{i=1}^{m} \lambda_i \cdot y^{\alpha_i}\right) dy\right] + \sum_{i=1}^{m} \lambda_i \cdot M_Y^{\alpha_i} \end{cases} \tag{4-12}$$

This optimization can be done in MATLAB using a simplex search method.

4.1.3 Verification example

A verification example has been done to make sure that the MATLAB script for M-DRM written for this study works properly. This example is from Zhang and Pandey (2013). In this study, the bending capacity of a reinforced beam is modelled as a function of six variables.

$$M_U(\boldsymbol{X}) = X_1 \cdot X_2 \cdot X_3 - \frac{X_1^2 \cdot X_2^2 \cdot X_4}{X_5 \cdot X_6} \quad (4\text{-}13)$$

Distributions of the random variables are listed in Table 4-1.

Table 4-1 Statistics of the variables.

Variable	**Description**	**Distribution**	**Units**	**Mean**	**COV**
X_1	Area of reinforcement	Lognormal	mm^2	1260	0.2
X_2	Yield stress of reinforcement	Lognormal	N/mm^2	300	0.2
X_3	Effective depth of reinforcement	Lognormal	mm^2	770	0.2
X_4	Stress–strain factor of concrete	Lognormal	–	0.35	0.1
X_5	Compressive strength of concrete	Weibull	N/mm^2	25	0.2
X_6	Width of beam	Normal	mm	200	0.2

Figure 4-2 compares the PDF of the bending capacity resulting from the code implemented in MATLAB with the results in Zhang and Pandey (2013). As can be seen, they are very close; the reported entropy in Zhang and Pandey (2013) is 5.9147, and in our work, is 5.9073.

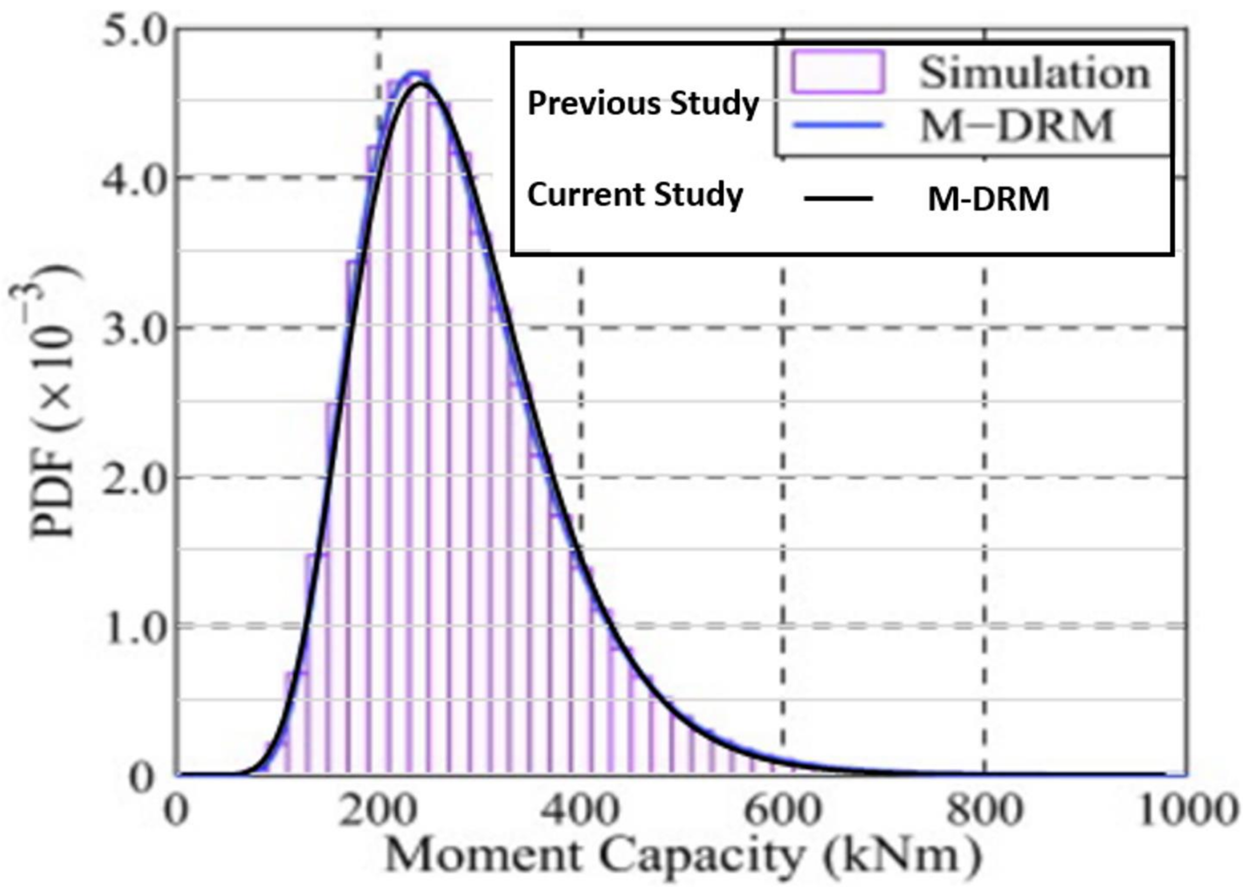


Figure 4-2 Comparing the results of the M-DRM code implemented in MATLAB with the results reported in Zhang and Pandey (2013).

4.2 Problem definition

This work aims to evaluate the CDF of the fretting fatigue life, N_f, of bridge stay cables at saddle supports and determine the sensitivity of the fretting fatigue life to the variable parameters in the problem. It should be noted that the analytical/closed-form equations were used to calculate the critical parameters including contact force/slip displacement. Therefore, based on the previous chapter, the first contact point is the critical point. Figure 4-3 shows a schematic view of a cable over a saddle. Details of the framework used for fretting fatigue analysis of a cable over a saddle are discussed in Chapter 3. Here, a summary of those methods is discussed.

As mentioned previously, a series of large-scale fretting fatigue tests were done at TU Berlin to evaluate the fretting fatigue life of cables at saddle supports. In this analysis, the test parameter, geometry, loading, and material properties of those tests were employed and finally, the results are compared with the results of the tests at TU Berlin.

Equation 4-14 was used to determine the fretting fatigue life, N_f of the cables. Based on this equation, the fretting fatigue life of the cable can be determined based on a damage parameter, called SWT, and the material properties of the cable.

$$SWT = \left(\sigma_f'\right)^2 \cdot \left(2 \cdot N_f\right)^{2 \cdot b} / E + \sigma_f' \cdot \varepsilon_f' \cdot \left(2 \cdot N_f\right)^{b+c} \qquad \text{4-14}$$

The output of this equation is the number of cycles to failure, N_f. The other parameters and their statistics are discussed in the following paragraphs.

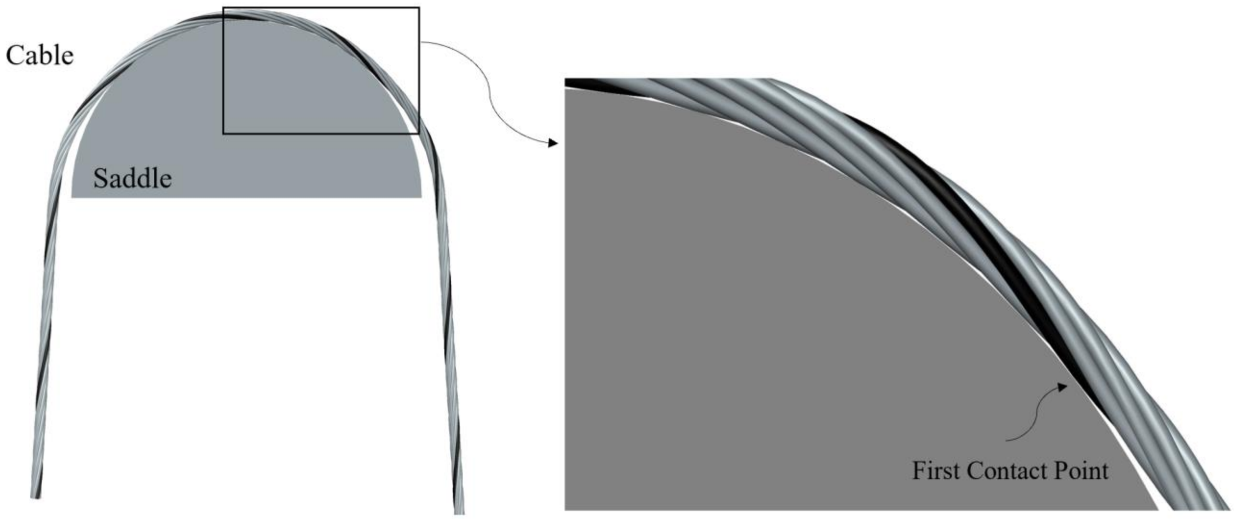


Figure 4-3 Schematic view of a cable over a saddle.

4.3 Analysis parameters and their statistics

4.3.1 SWT parameter

The SWT parameter is a damage parameter, which is based on the stresses and strains at the contact area of the cable and saddle. As the analysis time of a FE model of a full cable over a saddle is very long, only the contact point between the cable and the saddle is modelled. Figure 4-4 shows the 3D FE model of the contact point. The SWT parameter is determined based on the stresses and strains on the contact surface between the wire and saddle in this model.

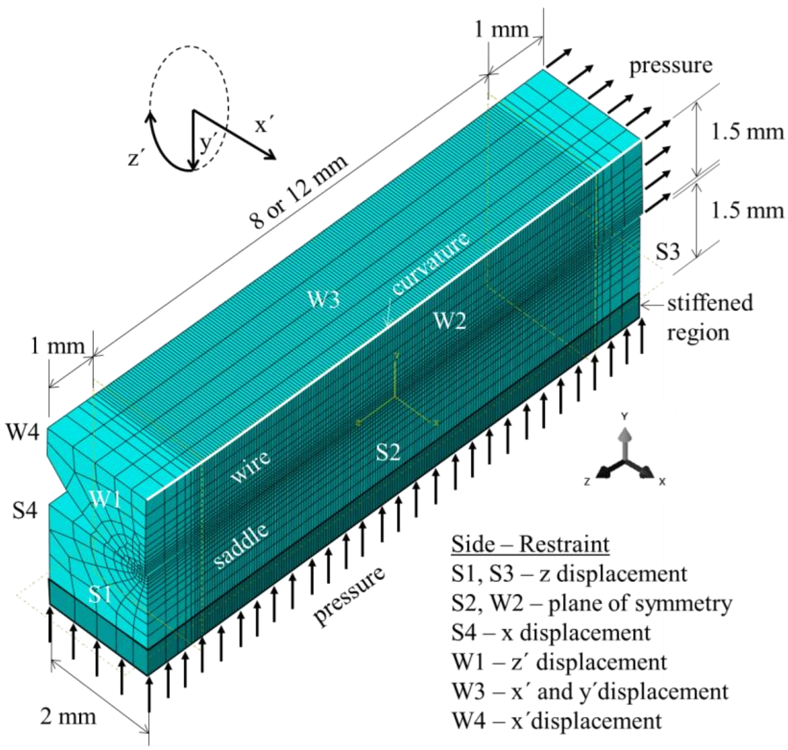


Figure 4-4 3D contact point model (Walbridge et al. 2017).

Slip displacement (relative displacement between the cable and the saddle) and contact force are two inputs of this FE model. These parameters can be calculated using the following equations:

$$\Delta = \frac{R}{E \cdot \mu \cdot A} \cdot \left(S_{max} \cdot \left(1 - \sqrt{\frac{S_{min}}{S_{max}}} \right) + S_{min} \cdot \left(1 - \sqrt{\frac{S_{max}}{S_{min}}} \right) \right) \qquad 4\text{-}15$$

$$F = \frac{S \cdot l}{R \cdot n} \qquad 4\text{-}16$$

where Δ is the slip displacement and F is the contact force. R is the saddle radii, A is the area of the cable, μ is the coefficient of friction, E is the elastic modulus of the cable, n is the number of outer wires of the cable, S_{max} is the maximum axial load of the cable, S_{min} is the minimum axial load of the cable, and l is the lay length of the cable. The geometry of the cable is shown in Figure 4-5. The equations developed for slip displacement and contact force (Equations 4-15 and 4-16) have not been proven to work perfectly. Also, the FE analysis in Chapter 3 shows that there are differences between the actual parameters and the results of these equations. One of the options in this situation is using bias factors to consider the uncertainty in these equations. A bias factor can be determined by comparing the experimental results and the results of an equation or analysis. However, a detailed large-scale experiment to measure these parameters could not be found in the literature. Therefore, based on the differences between the FE model and the discussed equations results, two bias factors were assumed: one for contact force, b_1, and another one for the slip displacement, b_2. A normal distribution with an average of 1 was assumed for both factors. A COV of 0.15 and 0.1 was assumed for b_1, and b_2 respectively as higher difference was seen between the results of contact force.

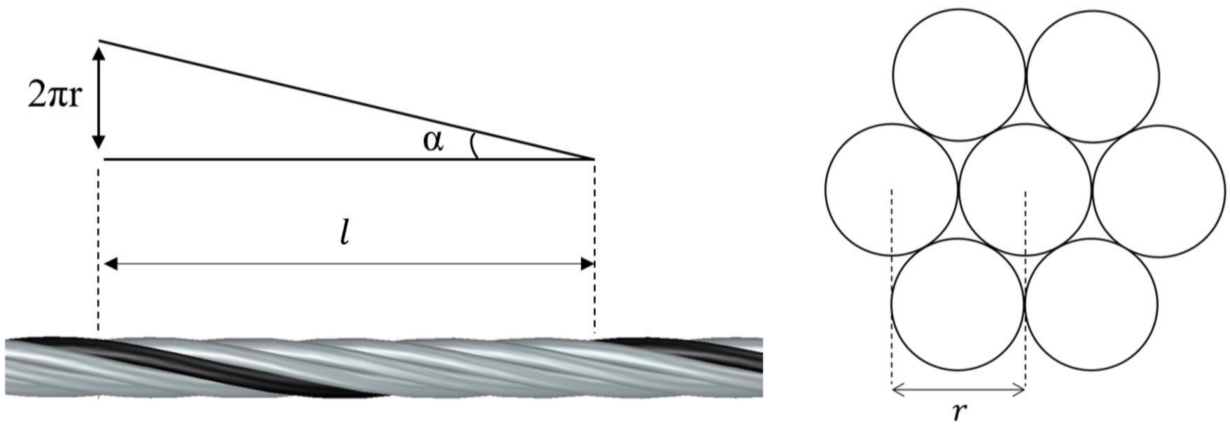


Figure 4-5 Geometric parameters of a cable.

The coefficient of friction is another input parameter of this FE model. Based on the tests done for the current study at the University of Waterloo, the COF ranges from 0.7 to 0.8. However, the results of tests done at TU Berlin show that the COF ranges from 0.6 to 0.7 (Mohareb 2020). Therefore, a uniform distribution between 0.6 to 0.8 was used for the current work.

4.3.2 Uncertainty in material properties

Elastic modulus is one of the parameters in this equation, which commonly is assumed to be constant. A constant value of 200 GPa was assumed in the current analysis.

Fatigue strength coefficient, σ_f', fatigue strength exponent, b, fatigue ductility coefficient, ε_f', and fatigue ductility exponent, c, are material properties related to the fatigue performance of materials. Based on (Zhu et al. 2017), b and c were assumed to be deterministic and a lognormal distribution with a COV of 0.05 and 0.16 was assumed for σ_f' and ε_f' respectively. The average values of the parameters were determined based on the fatigue tests in the current study using two different bridge cable types: galvanized and bare. It should be noted that the small COV that was assumed in this study for the fatigue strength coefficient might not be appropriate when the uncertainty in the material properties is very high. Therefore, considering a higher COV for these parameters might be of interest for design purposes. However, they will result in over-conservative results.

4.3.3 Summary of the parameters

The parameters of the current study are summarized in Table 4-2. The analysis was done based on two different wire types: galvanized and bare wire. Therefore, different values for the wires are listed in this table for material properties related to fatigue performance.

Table 4-2 Statistics of the parameters.

Parameter	Wire Type	Distribution	Average	COV
Number of outer wires, n	Bare/Galvanized	-	6	-
Radius of the saddle, R (mm)	Bare/Galvanized	-	1000	-
Lay length of the cable, l (mm)	Bare/Galvanized	-	216	-
The area of the cable, (mm^2)	Bare/Galvanized	-	150	-
Maximum axial load, S_{max} (kN)	Bare/Galvanized	-	126	-
Minimum axial load, S_{min} (kN)	Bare/Galvanized	-	96	-
Coefficient of friction, μ	Bare/Galvanized	Uniform [0.6-0.8]	0.7	0.082
Bias factor for contact force, b_1	Bare/Galvanized	Normal	1	0.15
Bias factor for slip displacement, b_2	Bare/Galvanized	Normal	1	0.10
Fatigue strength coefficient, σ_f' (MPa)	Bare	Lognormal	2675	0.05
Fatigue strength exponent, b	Bare	-	-0.0859	-
Fatigue ductility coefficient, ε_f'	Bare	Lognormal	0.2067	0.16
Fatigue ductility exponent, c	Bare	-	-0.5047	-
Fatigue strength coefficient, σ_f' (MPa)	Galvanized	Lognormal	2183	0.05
Fatigue strength exponent, b	Galvanized	-	-0.0657	-
Fatigue ductility coefficient, ε_f'	Galvanized	Lognormal	1.99	0.16
Fatigue ductility exponent, c	Galvanized	-	-0.8092	-

4.4 Probabilistic analysis frameworks

This section describes different MCS and M-DRM-based approaches or “frameworks” that can be applied to the fretting fatigue analysis of bridge stay cables, with their challenges and results. The simplest framework, completely based on MCS, would be as follows (see Figure 4-6):

- Step 1. Generate random values for bias factors and coefficient of friction from their distributions.
- Step 2. Determine contact force and slip displacement using Equations 4-15 and 4-16 and the trial values for bias factors and coefficient of friction.
- Step 3. Use FE analysis to determine the SWT parameter for the trial values of the contact parameters calculated in Step 2.
- Step 4. Generate random values for the material properties related to the fatigue performance of the wire (σ'_f, ε'_f).
- Step 5. Calculate the number of cycles to failure, N_f, using the SWT value obtained in Step 3 and material parameter trial values obtained in Step 4, by solving Equation 4-14.
- Step 6. Repeat Steps 1-5 until an accurate probability density function is obtained.

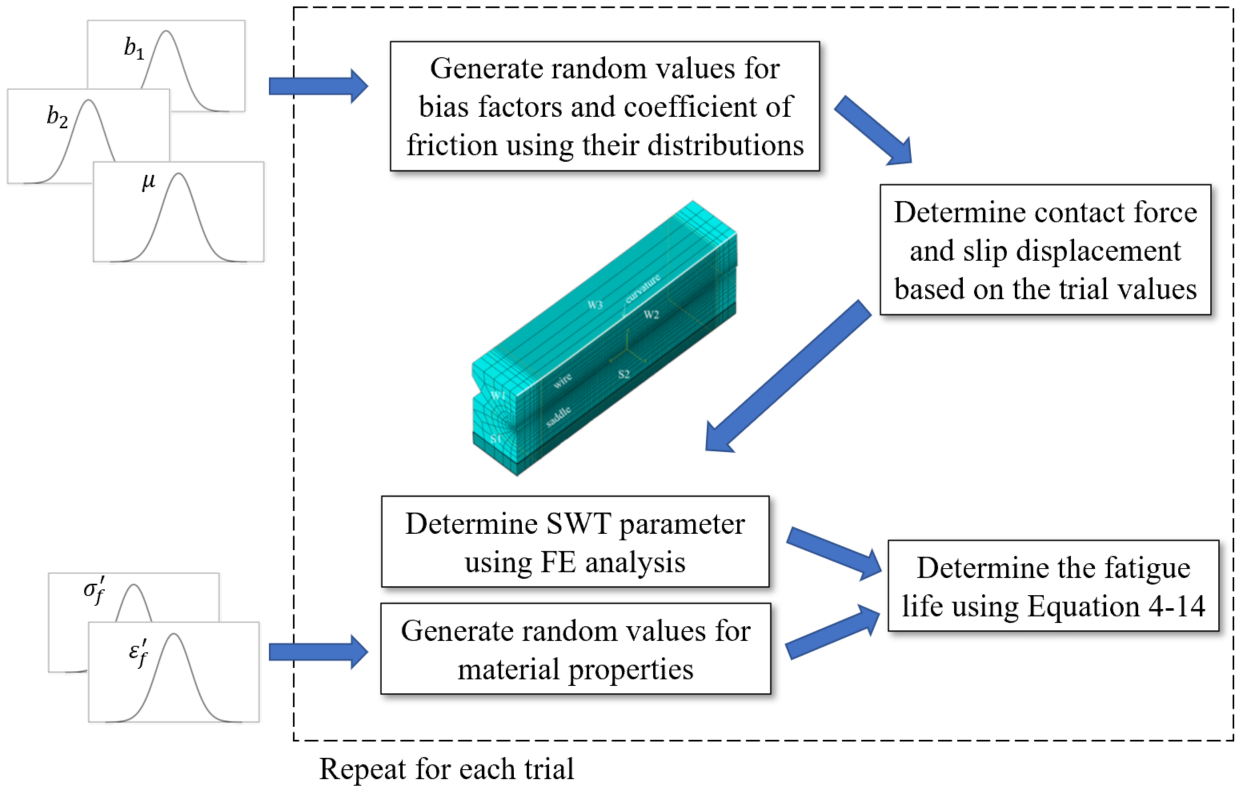


Figure 4-6 Probabilistic framework based on MCS.

This framework, however, is very inefficient. An FE analysis needs to be done for each trial, which lasts a few hours due to the small element size in the contact zone and the need to model the possible yielding of the saddle material. Therefore, it is not practical with current resources and analysis time. With this in mind, two alternative frameworks were developed to decrease the computational time related to the FE analysis step. These frameworks and their results are presented in detail in the following sections.

4.4.1 Probabilistic analysis using fretting maps

The first suggested framework, which avoids the FE analysis step for each trial, is based on interpolation and making use of the fretting maps generated in the previous chapter of this study. In fretting maps, the parameters that affect the FE analysis output, i.e. the peak value of the SWT parameter, are varied and the results are recorded. The effective parameters are coefficient of friction (COF), contact force, and slip displacement. Still, a number of FE analyses are required to generate the fretting maps. However, once these maps are established, probabilistic analyses can be performed with them for any problem where the contact point parameters are encompassed by the parameter ranges of the fretting maps. Afterwards, in the MCS, linear interpolation is used to find the desired trial value based on the inputs. The fretting maps employed for the current analysis are shown in Figure 4-7. These were generated in the previous chapter.

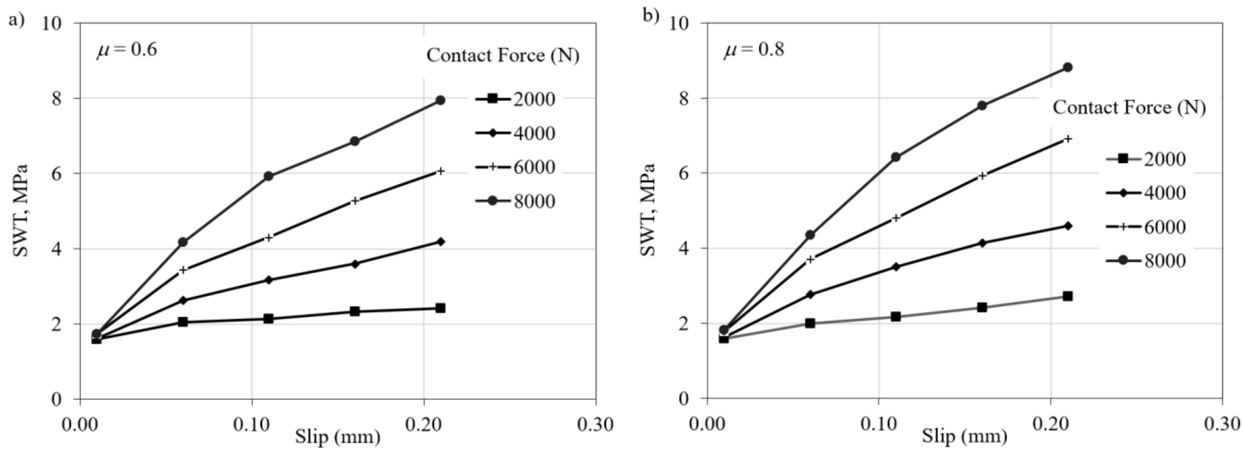


Figure 4-7 Fretting maps employed for the probabilistic analysis for COFs of 0.6 (a) and 0.8 (b).

The steps of an approach or framework for probabilistic analysis based on fretting maps and MCS are as follows and are shown in Figure 4-8:

- Step 1. Generate random values for bias factors and coefficient of friction from their distributions.
- Step 2. Determine contact force and slip displacement using Equations 4-15 and 4-16 and the trial values for bias factors and coefficient of friction.
- Step 3. Use the fretting maps shown in Figure 4-7, and interpolation (e.g., linear) to find the SWT parameter for trial values of the contact parameters calculated in Step 2.
- Step 4. Generate random values for the material properties related to the fatigue performance of the wire (σ'_f, ε'_f).
- Step 5. Calculate the number of cycles to failure, N_f, using the SWT value obtained in Step 3 and material parameter trial values obtained in Step 4, by solving Equation 4-14.
- Step 6. Repeat Steps 1-5 until an accurate probability density function is obtained.

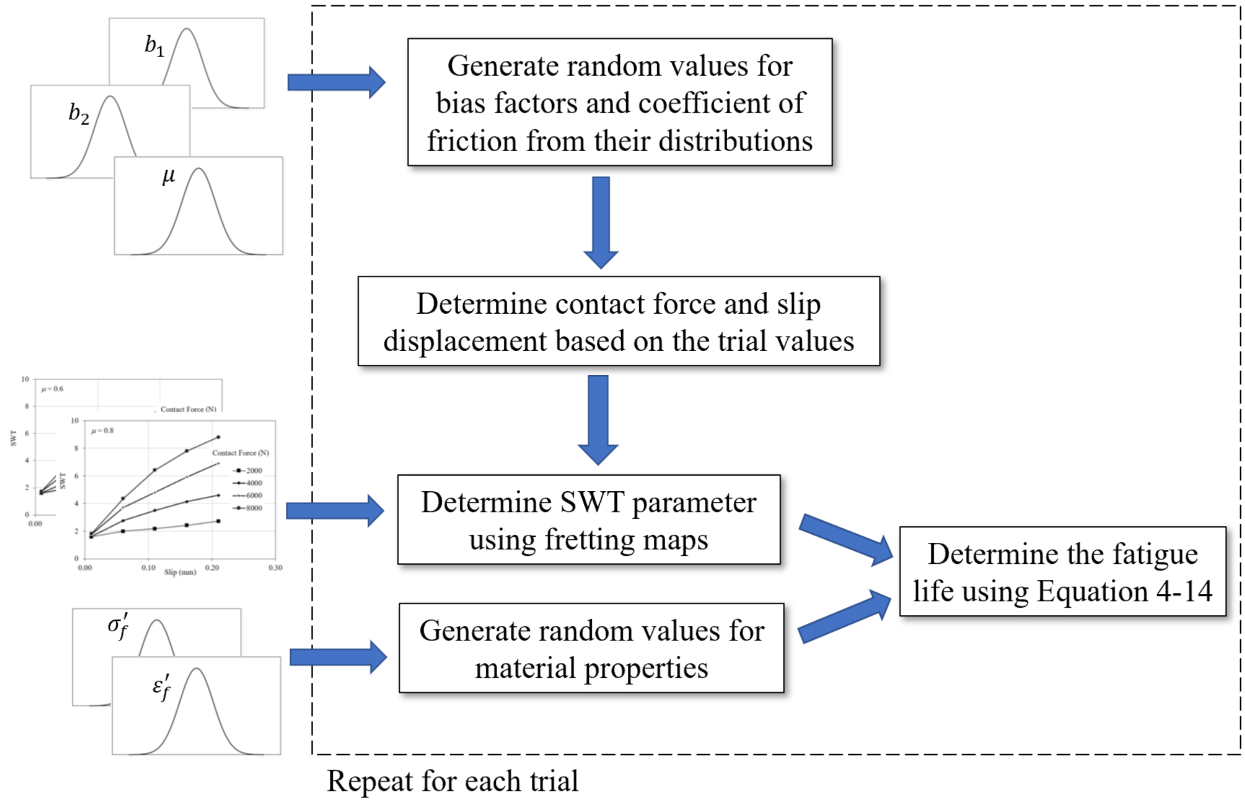


Figure 4-8 Probabilistic framework based on the fretting maps and MCS.

4.4.2 Probabilistic analysis using M-DRM

As mentioned earlier, while fretting maps can be used to enable probabilistic fatigue analysis at a contact point without an FE analysis required for each trial, similar fretting maps would have to be produced for a much wider range of conditions for this approach to be useful as a tool for general application to a broad range of design problems.

Another framework that could be used for this problem is based on M-DRM. In this framework, based on the input parameters, only a few FE analyses are required. For each analysis, one million trials were used to reach an acceptable level of accuracy at the tails of the CDFs. The steps of this framework are as follows and are shown in Figure 4-9:

- Step 1. Determine the input grid for the M-DRM based on the distribution of the variables.
- Step 2. Use FE analysis to determine the SWT parameter for the M-DRM trial values for the coefficient of friction, contact force, and slip displacement (based on X1, X2, X3 respectively in Table 4-3).

- Step 3. Determine the fatigue life, N_f, based on the SWT parameter and trial values for material properties: σ'_f, ε'_f (or X4 and X5 in Table 4-3) and Equation 4-14.
- Step 4. Use the M-DRM to determine the PDF of the fatigue life using the fatigue life of the trials in the input grid of M-DRM.

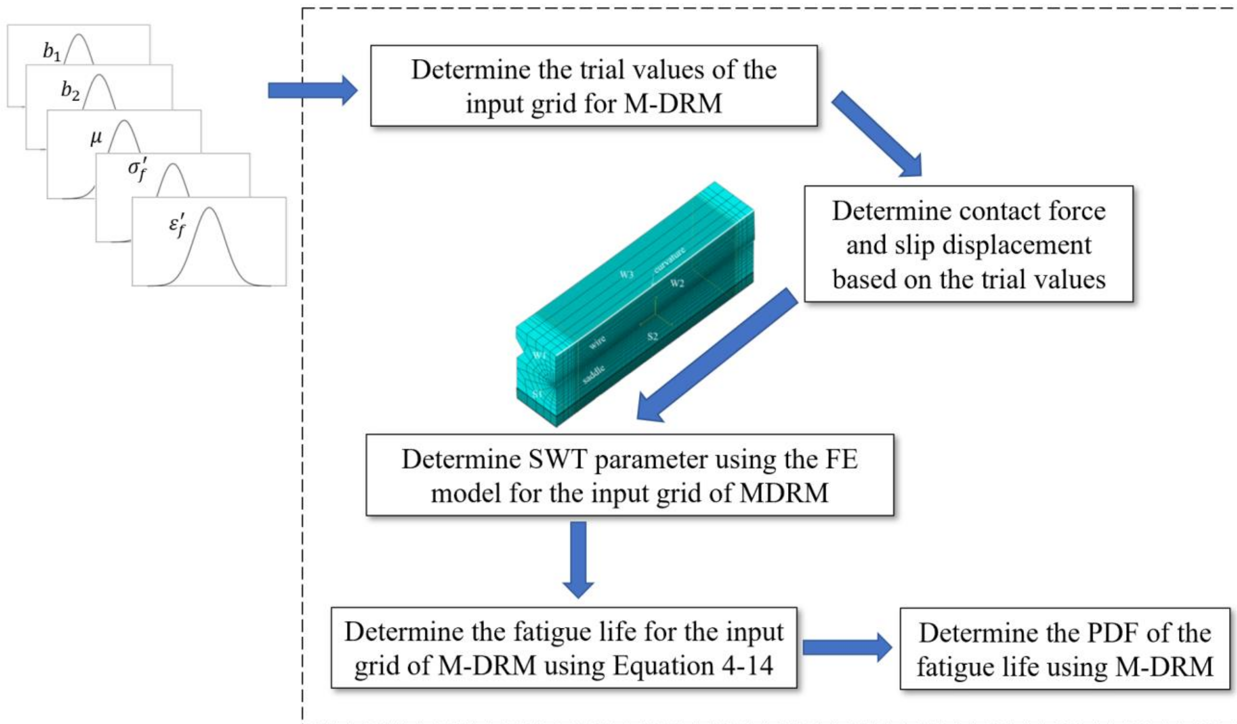


Figure 4-9 Probabilistic framework based on M-DRM.

The distributions of the random variables are the same as the ones used for MCS and are listed in Table 4-2. The analysis using M-DRM was done for both bare and galvanized cables. The input grid for these analyses is shown in Table 4-3 and Table 4-4. As can be seen in these tables, trial values for X1, X2, and X3 are the same in both tables as the material properties of the wire do not affect these parameters. It should be noted that the SWT parameter is also the same as it is the output of the FE analysis, which is a function of the first three variables (X1, X2, X3). However, the last two variables (X4 and X5) change based on the cable type.

Table 4-3 Input grid for M-DRM analysis based on the galvanized wire material properties.

Variable /Trial	*X1 (COF)*	*X2 (b1)*	*X3 (b2)*	*X4* ($\boldsymbol{\sigma'_f}$)	*X5* ($\boldsymbol{\varepsilon'_f}$)	*SWT*	$\boldsymbol{N_f}$	$\boldsymbol{Log}(\boldsymbol{N_f})$
1	0.6094	1	1	2183	1.99	3.449	1252562	6.098
2	0.6462	1	1	2183	1.99	3.485	1159118	6.064
3	0.7000	1	1	2183	1.99	3.465	1208293	6.082
4	0.7538	1	1	2183	1.99	3.366	1501633	6.177
5	0.7906	1	1	2183	1.99	3.440	1277632	6.106
6	0.7	0.5715	1	2183	1.99	2.304	26360717	7.421
7	0.7	0.7967	1	2183	1.99	2.981	3749936	6.574
8	0.7	1.0000	1	2183	1.99	3.465	1208293	6.082
9	0.7	1.2033	1	2183	1.99	4.137	326735	5.514
10	0.7	1.4285	1	2183	1.99	4.714	129110	5.111
11	0.7	1	0.7143	2183	1.99	3.215	2118132	6.326
12	0.7	1	0.8644	2183	1.99	3.324	1652011	6.218
13	0.7	1	1.0000	2183	1.99	3.465	1208293	6.082
14	0.7	1	1.1356	2183	1.99	3.625	863722	5.936
15	0.7	1	1.2857	2183	1.99	3.847	555918	5.745
16	0.7	1	1	1890.218	1.99	3.465	150555	5.178
17	0.7	1	1	2037.477	1.99	3.465	436480	5.640
18	0.7	1	1	2180.276	1.99	3.465	1185998	6.074
19	0.7	1	1	2333.084	1.99	3.465	3278757	6.516
20	0.7	1	1	2514.844	1.99	3.465	10202787	7.009
21	0.7	1	1	2183	1.24766	3.465	1197275	6.078
22	0.7	1	1	2183	1.58402	3.465	1202275	6.080
23	0.7	1	1	2183	1.96501	3.465	1207923	6.082
24	0.7	1	1	2183	2.43763	3.465	1214907	6.085
25	0.7	1	1	2183	3.09481	3.465	1224579	6.088

Table 4-4 Input grid for M-DRM analysis based on the bare wire material properties.

Variable /Trial	*X1 (COF)*	*X2 (b1)*	*X3 (b2)*	*X4* (σ'_f)	*X5* (ε'_f)	*SWT*	N_f	$Log(N_f)$
1	0.6094	1	1	2675	0.2067	3.449	533063	5.727
2	0.6462	1	1	2675	0.2067	3.485	504804	5.703
3	0.7000	1	1	2675	0.2067	3.465	519753	5.716
4	0.7538	1	1	2675	0.2067	3.366	605643	5.782
5	0.7906	1	1	2675	0.2067	3.440	540541	5.733
6	0.7	0.5715	1	2675	0.2067	2.304	4765054	6.678
7	0.7	0.7967	1	2675	0.2067	2.981	1159561	6.064
8	0.7	1.0000	1	2675	0.2067	3.465	519753	5.716
9	0.7	1.2033	1	2675	0.2067	4.137	208816	5.320
10	0.7	1.4285	1	2675	0.2067	4.714	109544	5.040
11	0.7	1	0.7143	2675	0.2067	3.215	772281	5.888
12	0.7	1	0.8644	2675	0.2067	3.324	647814	5.811
13	0.7	1	1.0000	2675	0.2067	3.465	519753	5.716
14	0.7	1	1.1356	2675	0.2067	3.625	410770	5.614
15	0.7	1	1.2857	2675	0.2067	3.847	302065	5.480
16	0.7	1	1	2316.232	0.2067	3.465	127785	5.106
17	0.7	1	1	2496.68	0.2067	3.465	260147	5.415
18	0.7	1	1	2671.663	0.2067	3.465	513137	5.710
19	0.7	1	1	2858.909	0.2067	3.465	1041888	6.018
20	0.7	1	1	3081.635	0.2067	3.465	2343391	6.370
21	0.7	1	1	2675	0.129593	3.465	474436	5.676
22	0.7	1	1	2675	0.16453	3.465	494949	5.695
23	0.7	1	1	2675	0.204104	3.465	518224	5.715
24	0.7	1	1	2675	0.253195	3.465	547167	5.738
25	0.7	1	1	2675	0.321456	3.465	587562	5.769

As discussed before, the general form of the PDF of the response using the M-DRM is:

$$\widehat{f_Y}(y) = exp\left(-\sum_{i=0}^{m} \lambda_i \cdot y^{\alpha_i}\right) \quad \text{4-17}$$

One of the parameters in this equation is m, or the number of terms in the equations. To find the optimal number of terms, the analysis can start with $m = 1$ and continue until the entropy does not decrease with increasing m. The output of the optimization in the M-DRM method is m factors (λ_i) and m corresponding exponent (α_i) and λ_0 (Based on Equations 4-11 and 4-12 earlier discussed in this chapter). The results for different numbers of m are summarized in Table 4-5 and Table 4-6 for galvanized and bare wires respectively. The analysis started with $m = 1$ and continued until the entropy converged. Looking at the entropies, it can be seen that three terms ($m = 3$) is adequate for both galvanized and bare wires. PDF and CDF results using different numbers of terms are compared in Figure 4-10 and Figure 4-11 for galvanized and bare wires respectively. A very small difference can be seen between the results with two to four terms. However, the results are significantly different when one term is used. The CDF results based on MCS and MDRM are compared in Figure 4-12 for the first contact point of saddle radius of 1000. The results are very close even for small CDF values on a logarithmic axis. It should be noted that apart from the interpolation error in the fretting maps, the maps were generated assuming the contact force does not change during the cyclic loading. However, M-DRM uses the exact solution and does not consider this simplifying assumption. Even with this assumption and interpolation errors, the results based on both methods are very close. All in all, M-DRM results can be considered to be more reliable as they are based on the exact solution of the FE model.

Table 4-5 M-DRM parameters for different numbers of terms (*m*) for galvanized wire.

Moments	Entropy	i	0	1	2	3	4
m=1	2.0823	λ_i	2.00136	1.9014E-12			
		α_i		13.2266			
m=2	0.7569	λ_i	268.4933	-197.4727	21.8353		
		α_i		0.4432	1.1411		
m=3	0.7501	λ_i	422.2582	77.9362	-289.1207	-115.5559	
		α_i		1.0180	0.1707	0.8312	
m=4	0.7434	λ_i	566.8728	-37.7001	49.7233	-472.9805	2.7504
		α_i		1.4708	1.3807	0.2117	1.7235

Table 4-6 M-DRM parameters for different numbers of terms (*m*) for bare wire.

Moments	Entropy	i	0	1	2	3	4
m=1	2.0010	λ_i	1.920922	5.5372E-12			
		α_i		13.2240			
m=2	0.3943	λ_i	371.9726	-245.1105	15.5693		
		α_i		0.4710	1.4198		
m=3	0.3864	λ_i	541.3486	86.3015	-341.7077	-134.6118	
		α_i		1.1418	0.2344	0.9110	
m=4	0.3840	λ_i	650.4716	2.7375	-330.4307	-37.4831	-208.5339
		α_i		2.1179	-0.0268	1.0363	0.0223

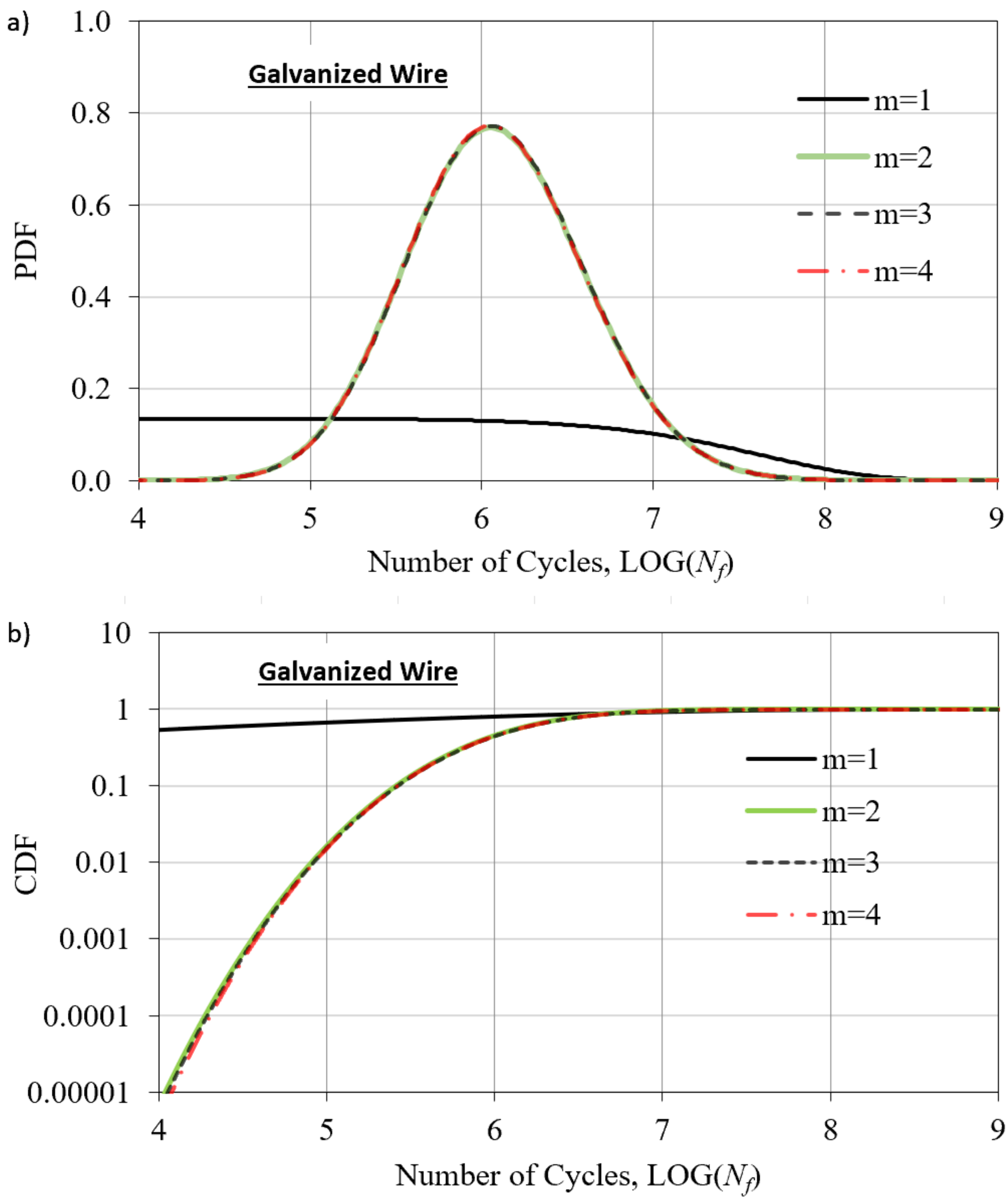
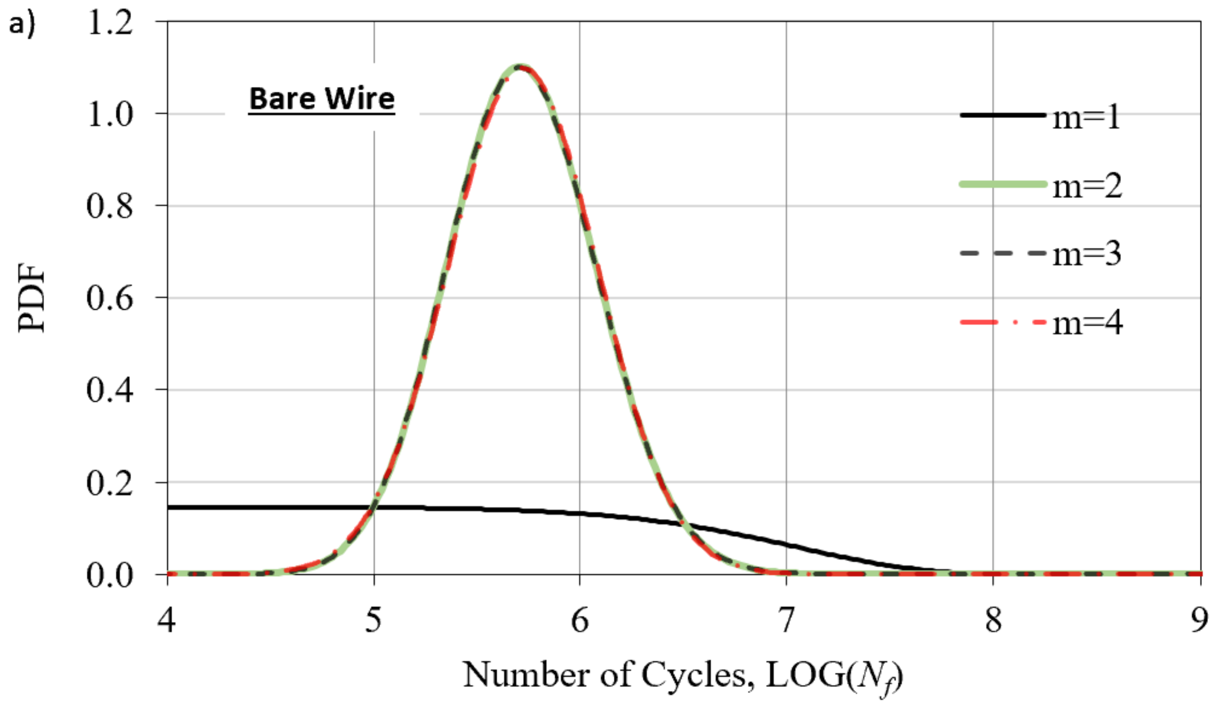


Figure 4-10 Comparing M-DRM results with different numbers of terms for galvanized wires.



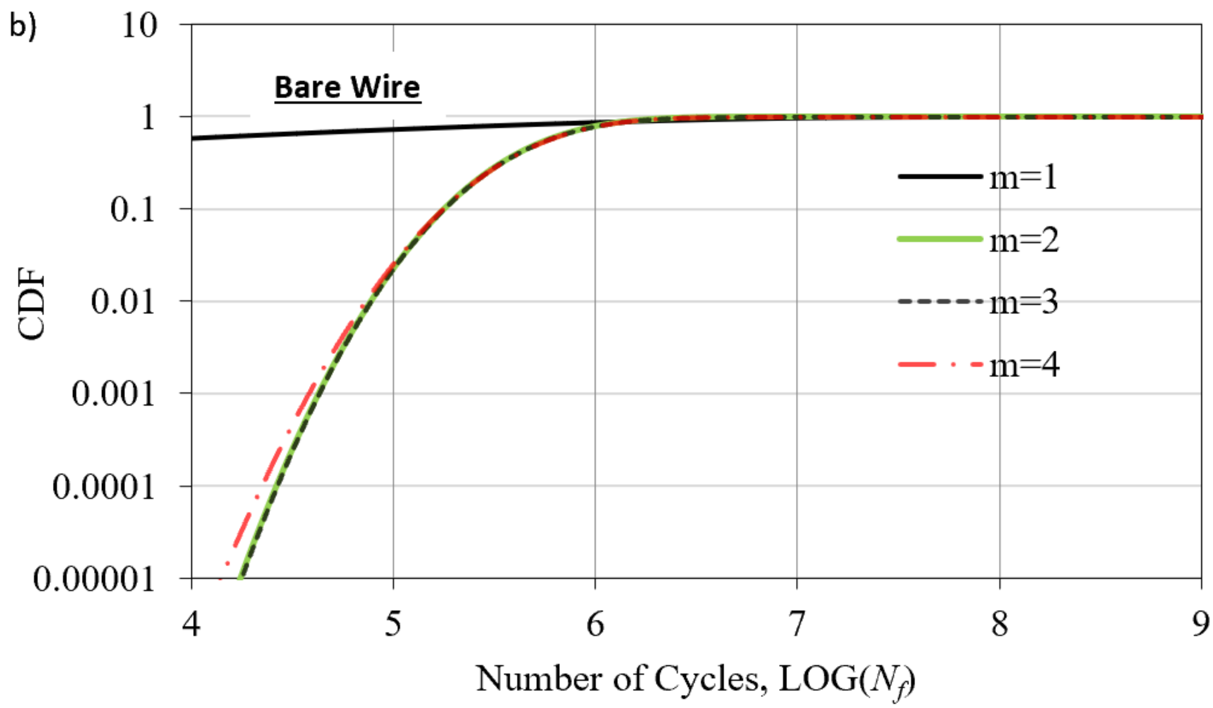


Figure 4-11 Comparing M-DRM results with different numbers of terms for bare wires.

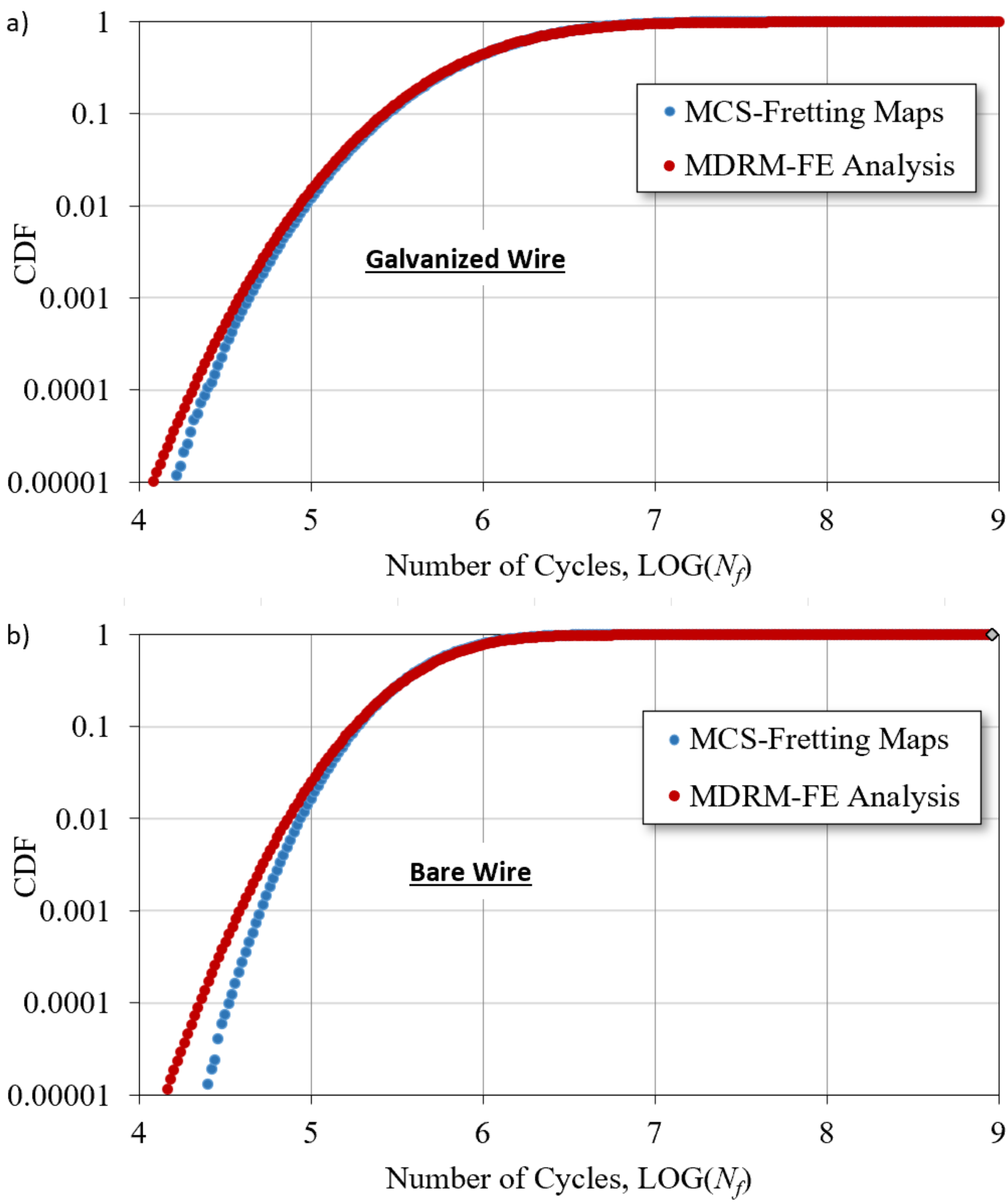


Figure 4-12 Comparing CDF results determined using the M-DRM and MCS-fretting maps on the logarithmic axis for galvanized (a) and bare (b) wires.

4.4.3 Sensitivity analysis using M-DRM

A sensitivity analysis was done using the M-DRM framework, to establish the relative importance of the input parameters. Based on Zhang and Pandey (2014), primary and total sensitivity index can be determined using Equations 4-18 and 4-19 respectively.

$$S_i \approx \frac{(\theta_i/\rho_i^2) - 1}{(\prod_{i=1}^{n} \theta_i/\rho_i^2) - 1} \tag{4-18}$$

$$S_{Ti} \approx \frac{1 - (\rho_i^2/\theta_i)}{1 - (\prod_{i=1}^{n} \rho_i^2/\theta_i)} \tag{4-19}$$

where ρ_i is the mean of each cut function, θ_i is the mean square of each cut function, S_i is the primary sensitivity factor, and S_{Ti} is the total sensitivity factor. The results of the sensitivity analysis are summarized in Table 4-7 and Table 4-8 for galvanized and bare wires, respectively. It can be seen that the difference between the primary and the total index is very small. Looking at the results, it can be seen that the bias factor of contact force (b_1) and the fatigue strength coefficient are the most important parameters. The sensitivity of the results to the bias factor for slip displacement (b_2), coefficient of friction, and fatigue ductility factor is limited. These sensitivity factors can be explained by looking at the fretting maps and Equation 4-14, which were used for calculating the fatigue life. First, looking at fretting maps shown in Figure 4-7, it can be seen that slight changes in slip displacement do not considerably change the SWT parameter as the SWT curves have small slopes at a fixed contact force, especially around the calculated values for contact force and slip displacement before applying the bias factor (which are around 4500 N and 0.096 mm respectively). But changes in contact force can considerably change the SWT parameter. Therefore, a higher sensitivity factor is expected for the contact force. Comparing the results for COFs of 0.6 and 0.8 does not show a considerable difference. Therefore, a small sensitivity factor is also expected for COF. Fatigue strength coefficient and fatigue ductility factor are two parameters in Equation 4-14 used for calculating the fatigue life. It should be noted that, in the long-life domain, the effect of the fatigue ductility factor is very limited, and the effect of the fatigue strength coefficient is significant. The fatigue life predictions in the current study are in the long-life region (typically more than 100,000 cycles). Therefore, high sensitivity of the results to the fatigue strength coefficient is expected.

Table 4-7 Sensitivity index results based on the galvanized wire material properties.

Variable	Parameter	S_i	S_{Ti}	$S_{Ti} - S_i$
X1	COF-Coefficient of friction	0.0058	0.0058	0.0000
X2	b_1-Bias factor for contact force	0.5722	0.5740	0.0018
X3	b_2-Bias factor for slip displacement	0.0393	0.0396	0.0003
X4	σ_f'-Fatigue strength coefficient	0.3808	0.3825	0.0017
X5	ε_f'-Fatigue ductility factor	0.0000	0.0000	0.0000
Sum		0.9981	1.0019	

Table 4-8 Sensitivity index results based on the bare wire material properties.

Variable	Parameter	S_i	S_{Ti}	$S_{Ti} - S_i$
X1	COF-Coefficient of friction	0.0053	0.0054	0.0000
X2	b_1-Bias factor for contact force	0.5815	0.5825	0.0010
X3	b_2 -Bias factor for slip displacement	0.0398	0.0400	0.0002
X4	σ_f'-Fatigue strength coefficient	0.3703	0.3713	0.0010
X5	ε_f'-Fatigue ductility factor	0.0020	0.0020	0.0000
Sum		0.9989	1.0011	

4.4.4 Analysis of all saddle radii using MDRM

Using this M-DRM approach to obtain the fretting fatigue life distribution, analyses were also performed for the first contact point of saddle radii of 500 and 1500 mm. The outcome is presented in Figure 4-13, where TU-Berlin test results are superimposed on M-DRM-derived curves associated with different survival probabilities (s.p.) based on the material properties of galvanized wires. Looking at this figure, it can be seen that the general trends are predicted reasonably well by the M-DRM model. In particular, it can be seen that the probabilistic model predicts the higher fatigue lives for the R = 500 and 1500 mm saddles. The 95% survival probability curve, which would be typically used in fatigue design for other structure types, such as welded structures, essentially represents a lower bound of the test data. Details on the MDRM analyses of saddle radii of 500 and 1500 mm including input grid tables, PDF and CDF results, and sensitivity analysis results are listed in Appendix B.

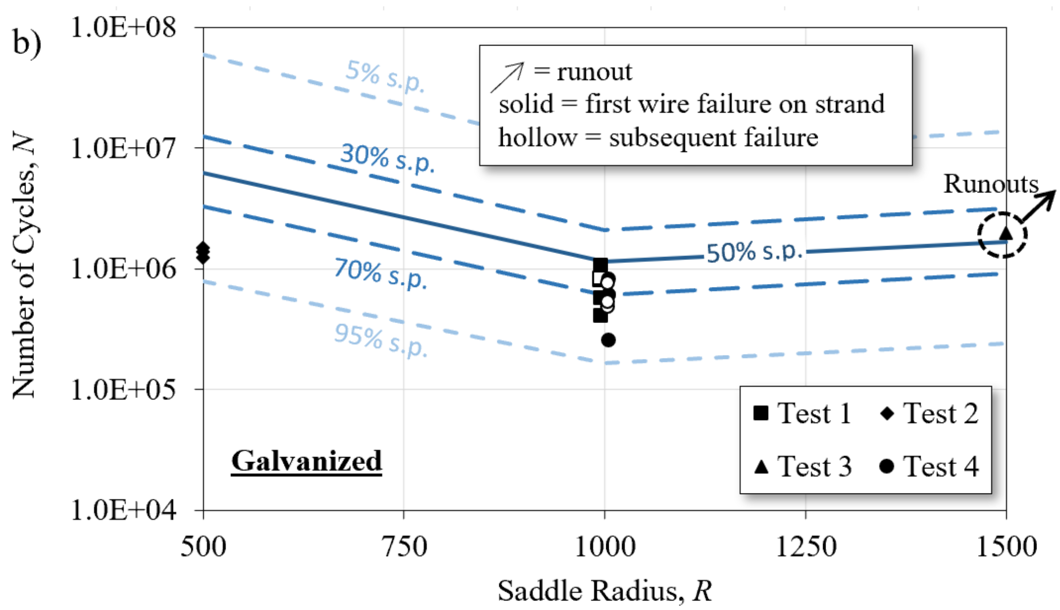


Figure 4-13 M-DRM analysis of pilot tests at TU Berlin.

5. Fretting Fatigue Tests of Bridge Stay Cable Wires

5.1 Background

Researchers have employed several full-scale and small-scale fretting fatigue test setups to evaluate the fretting fatigue behaviour of components. Full-scale tests of some components can be very time-consuming and costly. However, alternative small-scale tests, while requiring care to be taken in the treatment and simplification of certain aspects such as the boundary conditions, are often found to be much more efficient and economical. Therefore, alternative small-scale tests are very popular in different engineering fields. In small-scale fretting tests, typically the specimen is fixed at one end and cycled between maximum and minimum stresses at the other end. A fretting setup is required to apply the contact force to the specimen. One traditional method uses floating bridge-type pads (see Figure 5-1) and a ring to transfer the contact force to the specimen (Majzoobi et al. 2007, Majzoobi et al. 2009). However, controlling the test parameters is challenging with these setups. To overcome the issues with floating pads, an alternative approach with fixed pads has been adopted in many studies. In these setups, several methods can be used to apply the contact force: employing hydraulic actuators, tightening threaded rods, and using weights (Szolwinski et al. 1998, Murthy et al. 2006, Hojjati-Talemi et al. 2014, Guo et al. 2020).

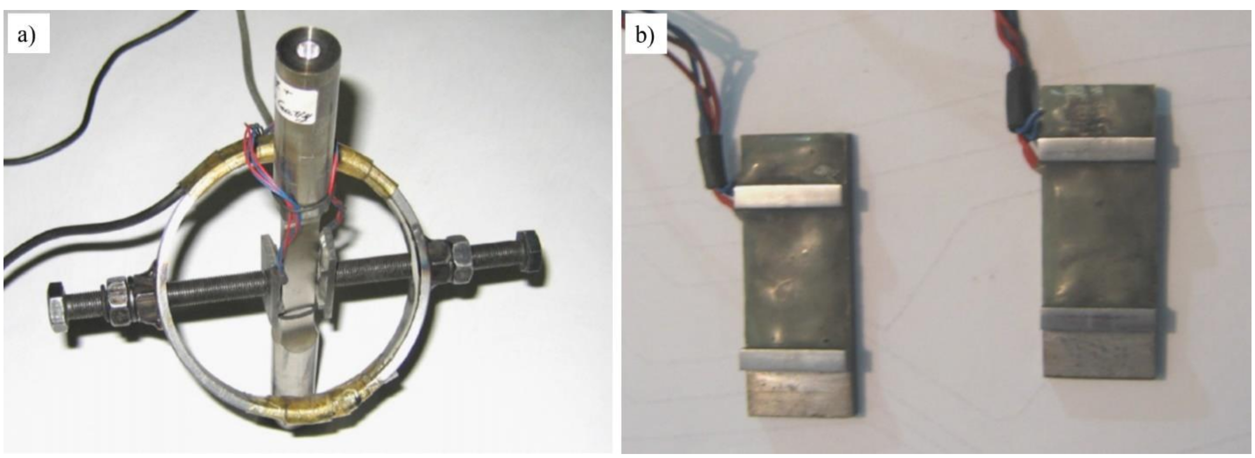


Figure 5-1 Fretting fatigue test setup using a ring (a), floating bridge-type pads (b) (Majzoobi et al. 2009).

5.2 Fretting fatigue test setup

Figure 5-2 shows the test setup designed for the current thesis for use with an MTS testing frame. Two plates with four round bars were used to elevate the fretting setup and put it over the bottom grip of the MTS machine. The wire specimen is fixed at the top grip and cycled between the maximum and minimum stress with the bottom grip. Load cells at the top and bottom grips monitor the axial load of the wire at the top and the bottom. The recorded load cell readings were used to determine the frictional force and slip displacement at the contact point.

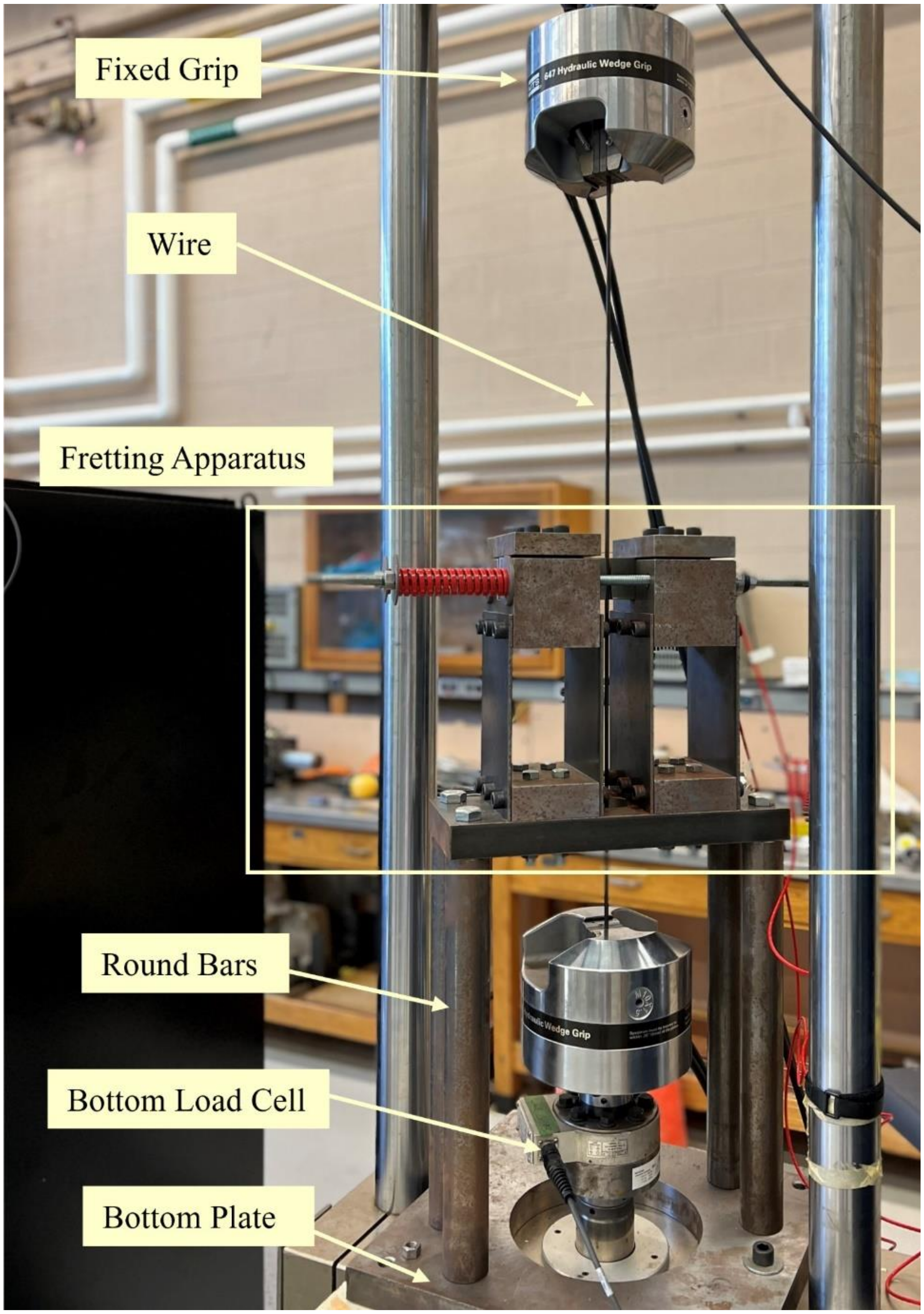


Figure 5-2 Fretting fatigue test setup designed for the current study.

Details of the fretting apparatus are shown in Figure 5-3. This setup is based on fixed pads and the idea of employing threaded rods to apply the contact force to the specimens. The contacting pads are clamped to the holding boxes using four bolts; the contacting pads get replaced after each fretting fatigue test. The contact force is applied to the wire by tightening two threaded rods that go through the boxes. The contact force is measured using two washer load cells, one on each threaded rod. One spring is used on each threaded rod to increase the accuracy of loading and decrease the loss of the contact force during the tests. Two flexible plates are used to hold each box. These plates should be sufficiently stiff in the vertical direction but flexible in the horizontal direction to ensure they do not attract a significant portion of the applied lateral force with a small amount of wear or plastic deformation of the contact pads during the tests.

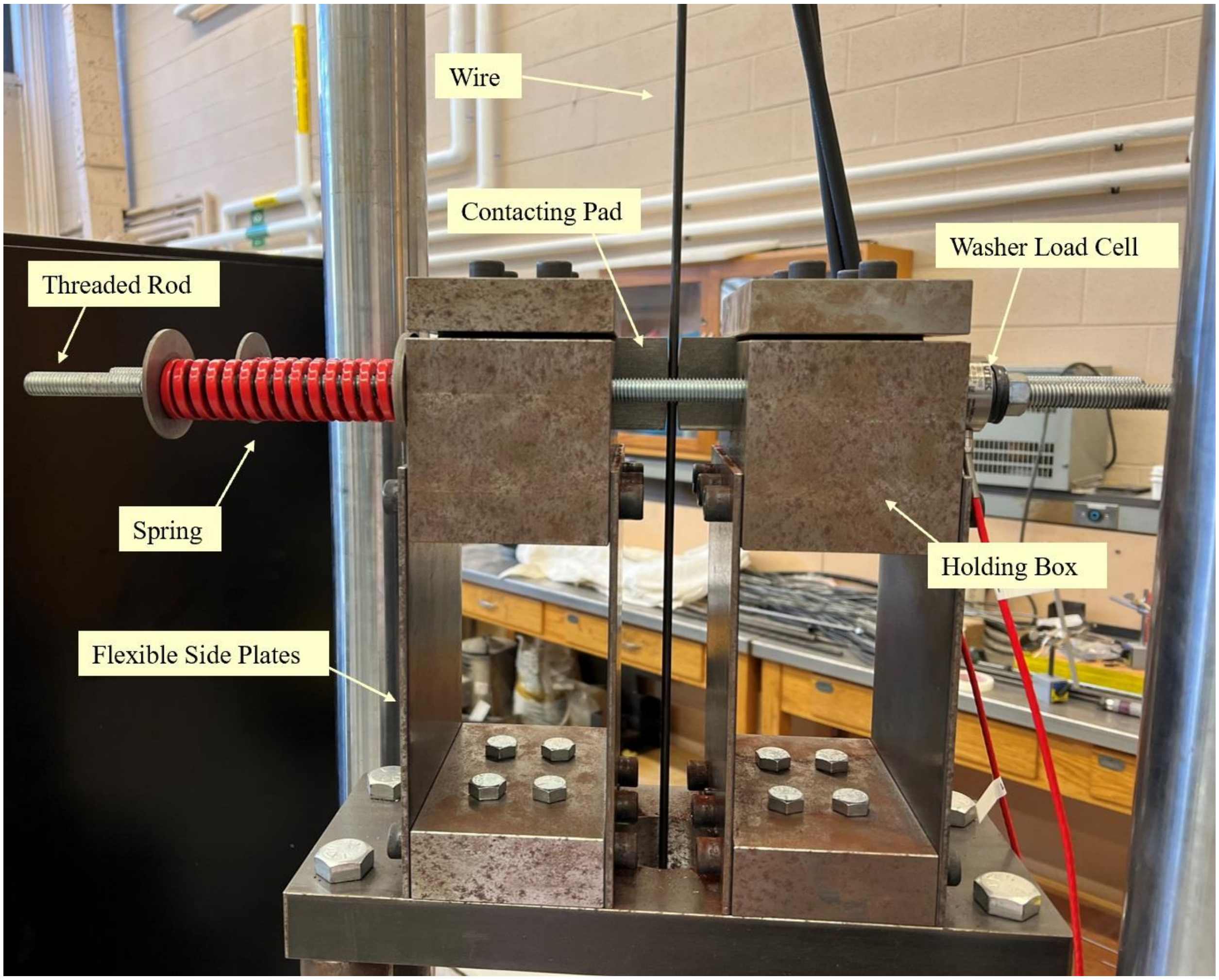


Figure 5-3 Details of the fretting apparatus.

Typically, dog bone specimens are used in fretting fatigue or fatigue tests to avoid failures at the grips. In this work, however, the whole wire without any machining was employed so that effects of the wire surface treatment could be studied. With this decision, two points were found to be critical for fatigue failure: the contact point and the bottom grip where the cyclic load is applied. The upper grip was not critical as the load range at the top was much lower than the load range at the bottom grip, due to the frictional force at the contact point. To decrease the possibility of failure at the bottom grip, custom inserts with a notch having the same radius as the wires were designed, fabricated, and used for the fretting fatigue tests (Figure 5-4).

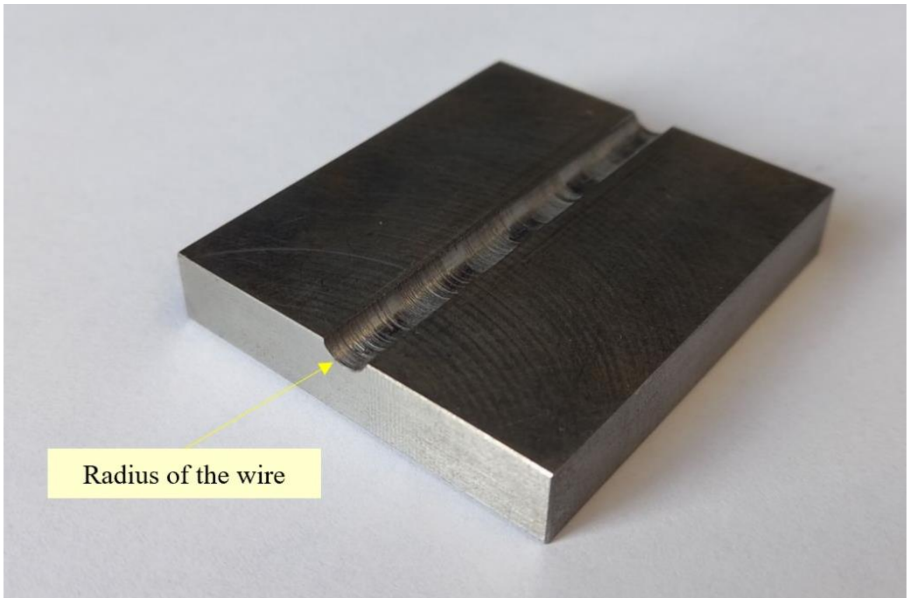


Figure 5-4 Custom inserts for the wire specimens.

5.3 Fretting fatigue tests parameters

5.3.1 Test materials

Two steel cable wire types – one with and one without surface galvanization – were studied in the current work (see Figure 5-5 (a)). The factory reported guaranteed ultimate tensile strength, GUTS, of the wires was 1860 MPa. Two contacting pads were used for each test. The pad contact surface has a curvature equal to the curvature of the outer wire of the cable due to it being twisted about the central wire. The geometry of the pads is shown in Figure 5-5. The pads were machined from mild steel 0.75×2 inch rectangle bars with a hardness of 89 HRB. Curved contact surfaces were machined at both ends of the pad so that each pad could be used for two tests.

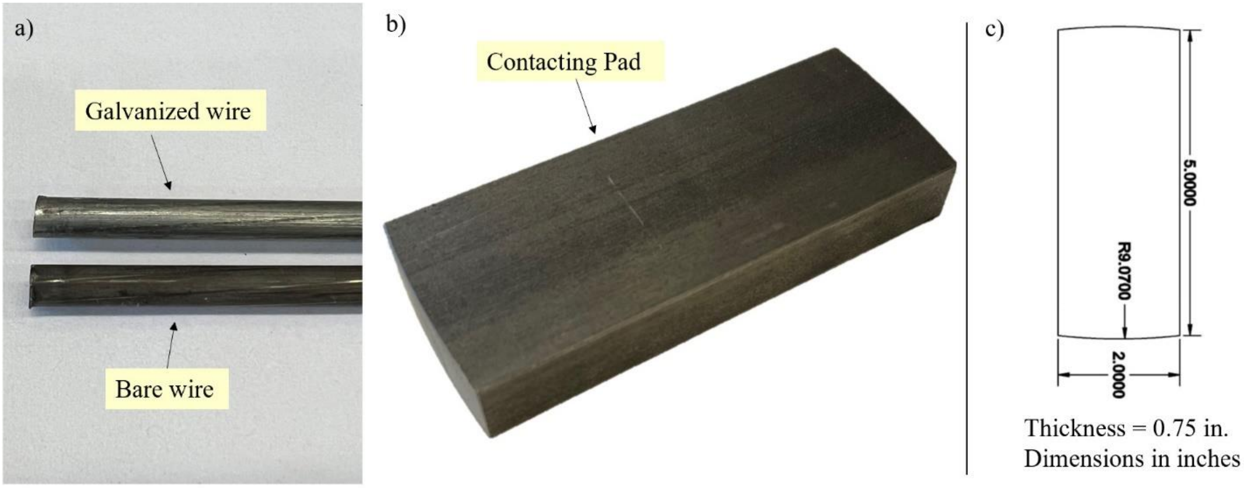


Figure 5-5 Studied wires (a), contacting pad (b), the geometry of the pad (c).

5.3.2 Loading

Based on the criteria specified in fib (2005), the cable wires should be cycled between the maximum stress of 45% GUTS (837 MPa) and minimum stress (637 MPa) to result in a stress range of 200 MPa (an R ratio of 637/837 = 0.76). Apart from this design stress range, a few tests were done at a stress range of 300 MPa to investigate the effect of stress range on fretting fatigue performance (an R ratio of 537/837 = 0.64). Several contact forces ranging from 750 to 6000 N were employed in this work. The wires were first loaded to the minimum stress, then the contact force was applied to the wire and, finally, the wire was cycled between the maximum and minimum stresses at a frequency of 15 Hz. A real bridge cable is subjected to variable amplitude loading, and there is a lower frequency for high stress ranges in bridge structures. In this work, however, given the time limits of the project, a higher frequency was employed. During the tests, no temperature raise was observed on the wires in stick-slip regimes as a result of high frequency since the cables do not completely move over the pads.

5.4 Fretting fatigue test results

Figure 5-6 shows two contact surfaces after testing for two typical fretting regimes in fretting fatigue tests: Gross sliding and stick-slip regimes. The gross sliding regime was seen at low contact forces of 750 and 1100 N. The amount of wear was significant in this regime; however, no wire failed in this fretting condition. The stick-slip regime was observed at higher contact forces with a limited amount of wear. All of the observed failures were at the stick-slip regime.

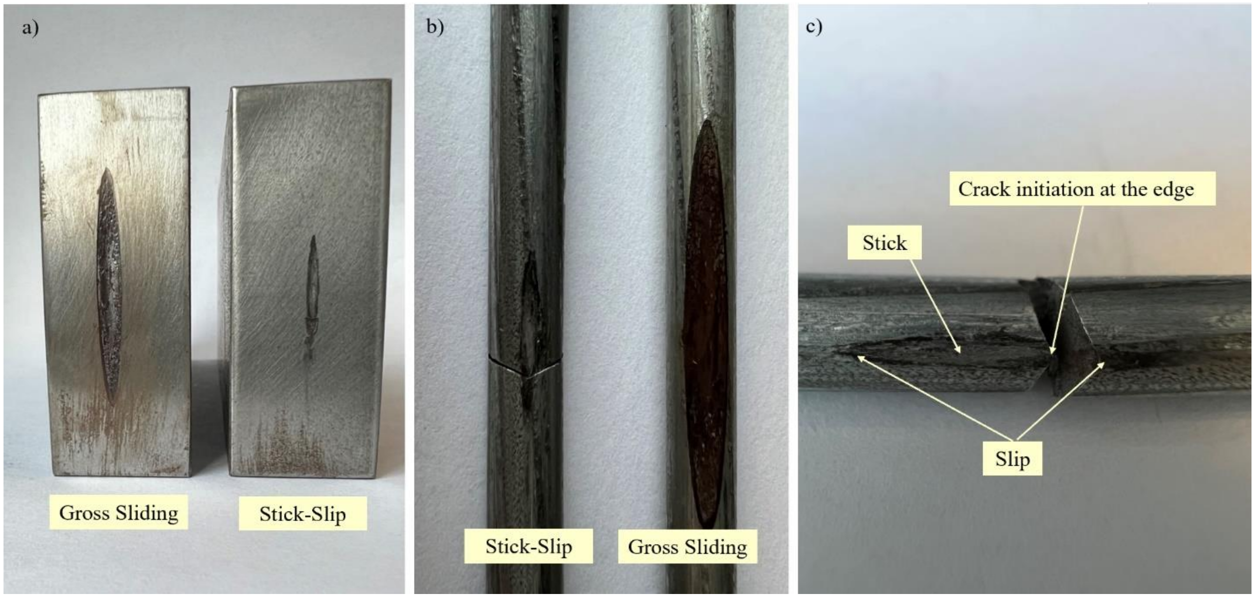


Figure 5-6 Typical wear on pads (a) and wires (b) and crack initiation location at the edge of the contact area (c).

Figure 5-7 shows the fretting fatigue test results for the galvanized wires. Looking at the results for the stress range of 200 MPa, it can be seen that the fretting fatigue life first decreases with an increase in the contact force, then at a critical contact force, 3000 N, the fretting fatigue life starts to increase. The same trend was seen for the results at 300 MPa with a critical contact force at 4500 N; however, the difference between the results at 300 MPa was relatively small. Comparing the results for the two studied stress ranges shows the significant effect of the stress range on the fretting fatigue life of the cables. The decrease in fatigue life with an increase in the stress range is a typical trend in fretting fatigue tests and fatigue tests in general.

Surprisingly, no failure was observed when the bare wires were used in the tests with the same parameters (see Figure 5-8). First, the bare wires were tested at the design stress range, 200 MPa, and contact forces of 1500 and 3000 N which were the critical contact forces for the galvanized wire at this stress range. No bare wire failed with these sets of parameters while all the galvanized wires failed with these sets of parameters. Then, two tests were done at the higher stress range of 300 MPa and contact forces of 4500 and 6000 N. The bare wires in these tests failed at the bottom grip instead of the fretting point. However, all six galvanized wire specimens failed at the fretting point below 200,000 cycles with these test parameters. Further efforts to vary the loading parameters to cause fretting fatigue failures in the bare wires were hampered by failures at the grips, suggesting a different way of gripping the specimens would be required to create conditions for fretting failure.

Based on (fib 2005, PTI 2012) a run-out limit of 2 million cycles should be used. In this work, however, failures between 2 and 5 million cycles were observed. Therefore, a run-out limit of 5 million cycles was employed.

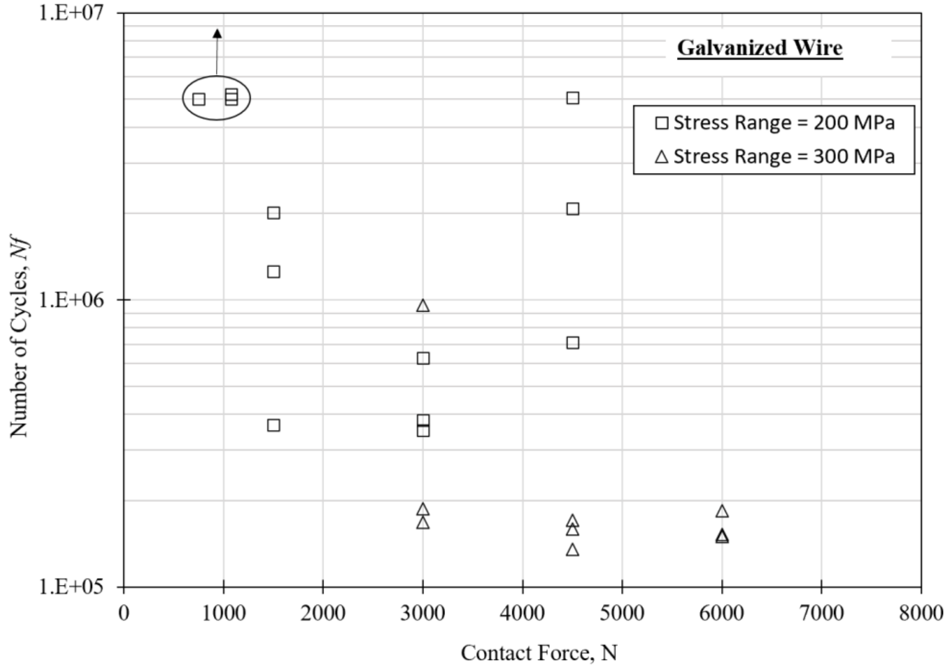


Figure 5-7 Fretting fatigue life of the galvanized wire.

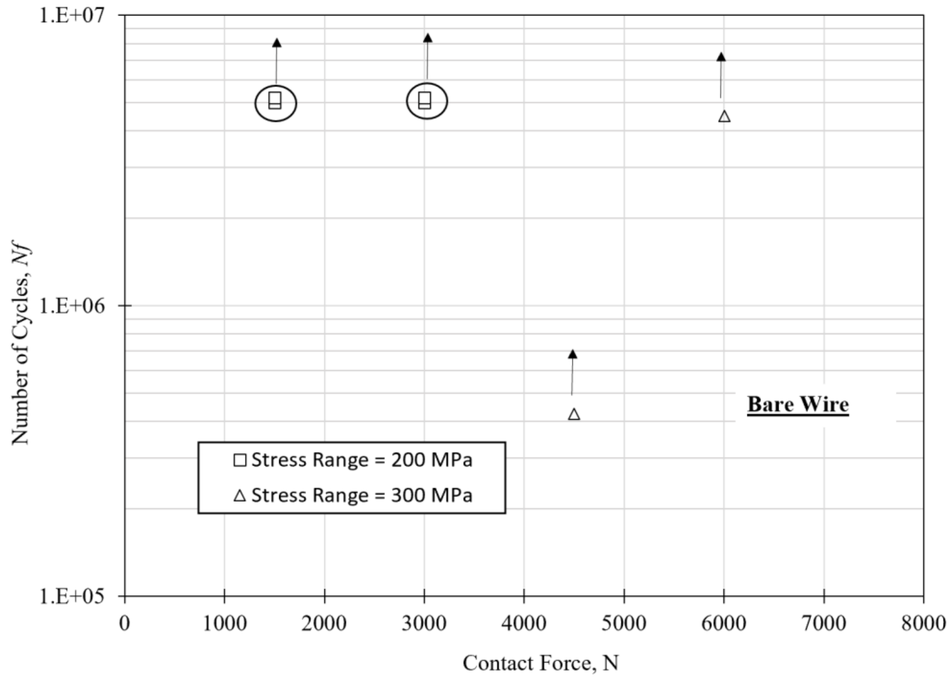


Figure 5-8 Fretting fatigue life of the bare wire.

This significant difference between the performance of these two wires raised questions regarding possible differences in material properties or the microstructure of the wires. The material properties and microstructure of the wires are compared later in this chapter in an attempt to answer these questions. Details of these tests are summarized in Table 5 1. Details on the calculation of slip displacement and COF are discussed in the following paragraphs.

Table 5-1 Summary of fretting fatigue test results (+ indicates “runout”).

Test #	Wire Type	Stress Range (MPa)	Maximum Stress (MPa)	Minimum Stress (MPa)	Contact Force (N)	Quasi-COF	Slip Displacement (mm)	Fretting Fatigue Life (N_f)
1	Galvanized	200	837	637	750	-	0.3555	+5000000
2	Galvanized	200	837	637	1100	-	0.2504	+5000000
3	Galvanized	200	837	637	1100	-	0.2391	+5000000
4	Galvanized	200	837	637	1500	0.6757	0.0408	2011238
5	Galvanized	200	837	637	1500	0.6639	0.0410	367221
6	Galvanized	200	837	637	1500	0.6689	0.0390	1254930
7	Galvanized	200	837	637	3000	0.3383	0.0392	381503
8	Galvanized	200	837	637	3000	0.3363	0.0392	351020
9	Galvanized	200	837	637	3000	0.3415	0.0381	627301
10	Galvanized	200	837	637	4500	0.2277	0.0381	711992
11	Galvanized	200	837	637	4500	0.2290	0.0382	2078948
12	Galvanized	200	837	637	4500	0.2293	0.0368	5072193
13	Galvanized	300	837	537	3000	0.5090	0.0587	958406
14	Galvanized	300	837	537	3000	0.5288	0.0571	186643
15	Galvanized	300	837	537	3000	0.5128	0.0575	167805
16	Galvanized	300	837	537	4500	0.3473	0.0551	158661
17	Galvanized	300	837	537	4500	0.3445	0.0557	170981
18	Galvanized	300	837	537	4500	0.3434	0.0568	135249
19	Galvanized	300	837	537	6000	0.2616	0.0547	184444
20	Galvanized	300	837	537	6000	0.2608	0.0548	149746
21	Galvanized	300	837	537	6000	0.2612	0.0552	152615
22	Bare	200	837	637	1500	0.6330	0.0413	+5000000
23	Bare	200	837	637	1500	0.6380	0.0414	+5000000
24	Bare	200	837	637	3000	0.3230	0.0355	+5000000
25	Bare	200	837	637	3000	0.3260	0.0359	+5000000
26	Bare	300	837	537	4500	0.3259	0.0519	+424916
27	Bare	300	837	537	6000	0.2470	0.0512	+4513843

5.5 Coefficient of friction

The coefficient of friction, COF, is a key parameter in fretting fatigue problems. Determining this parameter in fretting fatigue tests has always been a challenging issue. Several works have shown that the COF typically increases during the fretting fatigue tests due to surface modification and wear (Hills et al. 1988, McColl et al. 2004, Jin and Mall 2004).

In the gross sliding regime, sliding occurs all over the contact surface, and average COF can be defined as the ratio of frictional force to the contact force (*Q*/*P*).

$$COF_{ave\ gross\ sliding} = \frac{Q}{P} \tag{5-1}$$

However, in the stick-slip regime, this ratio (*Q*/*P*) is only the normalized frictional force or a quasi-COF, as defined in McColl et al. (2004). In this regime, the center of the contact has a stick regime with a low volume of wear and the borders of the contact are slipping and have a higher amount of wear. Therefore, each point can have a different COF. The average COF in this regime falls between quasi-COF (*Q*/*P*) and the COF of the gross sliding regime. The average value for the COF can only be determined if the wire completely moves over the pad, which does not happen during the tests in this regime.

$$\frac{Q}{P} < COF_{ave\ stick-slip} < COF_{ave\ Gross\ Sliding} \tag{5-2}$$

Figure 5-9 shows quasi-COF (*Q*/*P*) results for the tests in the stick-slip regime (contact forces ≥ 1500 N). It can be seen in this figure that this parameter decreases with an increase in the contact force or a decrease in stress range. This trend has been seen in other studies in the literature (McColl et al. 2004, Jin and Mall 2004).

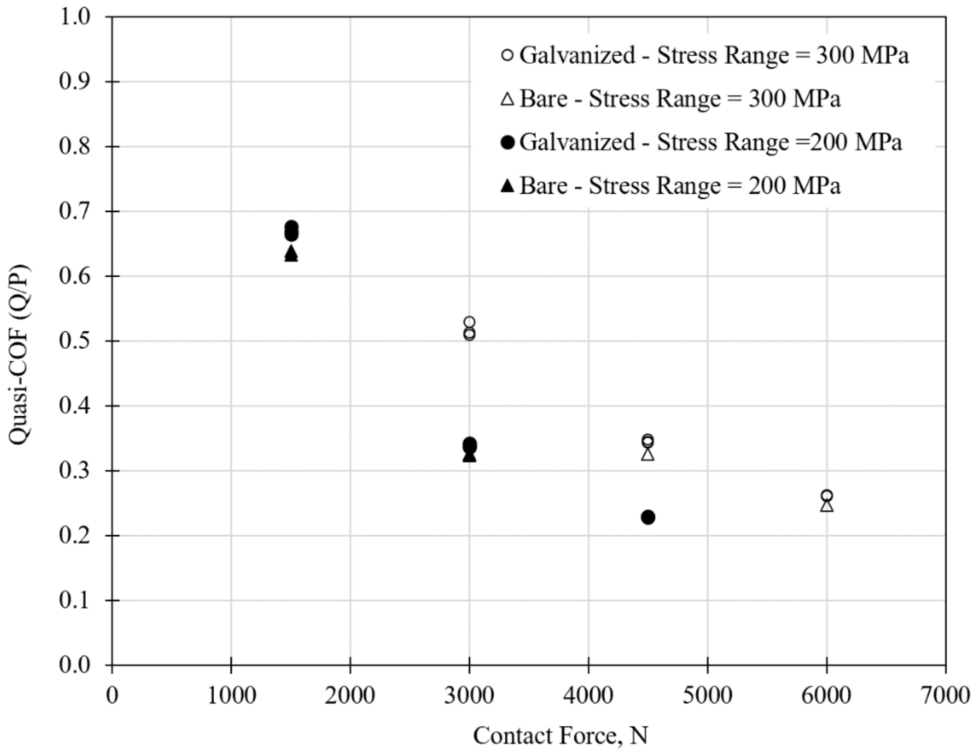


Figure 5-9 Quasi-COF results for the tests in the stick-slip regime.

To evaluate the COF in the gross sliding regime, two fretting fatigue tests with completely similar conditions were carried out using a bare and a galvanized wire. An axial load range of 4.3 kN and a contact force of 1 kN was employed for these tests. Figure 5-10 shows the wear on the wires after the friction tests. Figure 5-11 compares the COF measurements versus the number of cycles for the studied wires. As can be seen in the figure, the COF increases during the first cycles and then stabilized around a maximum value of ~0.75.

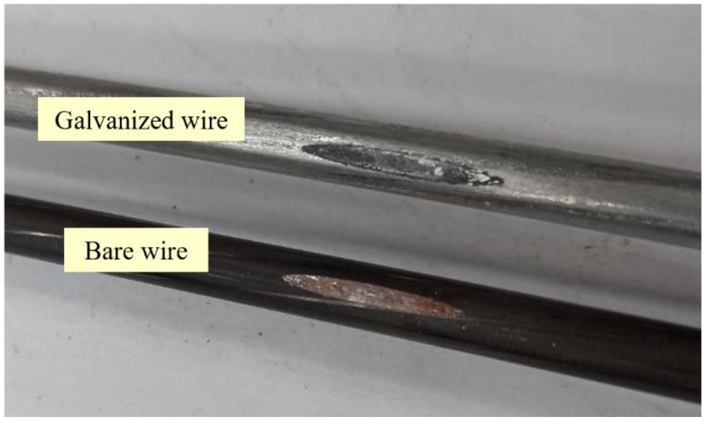


Figure 5-10 Wear on the bare and the galvanized wire after friction tests in the gross sliding regime.

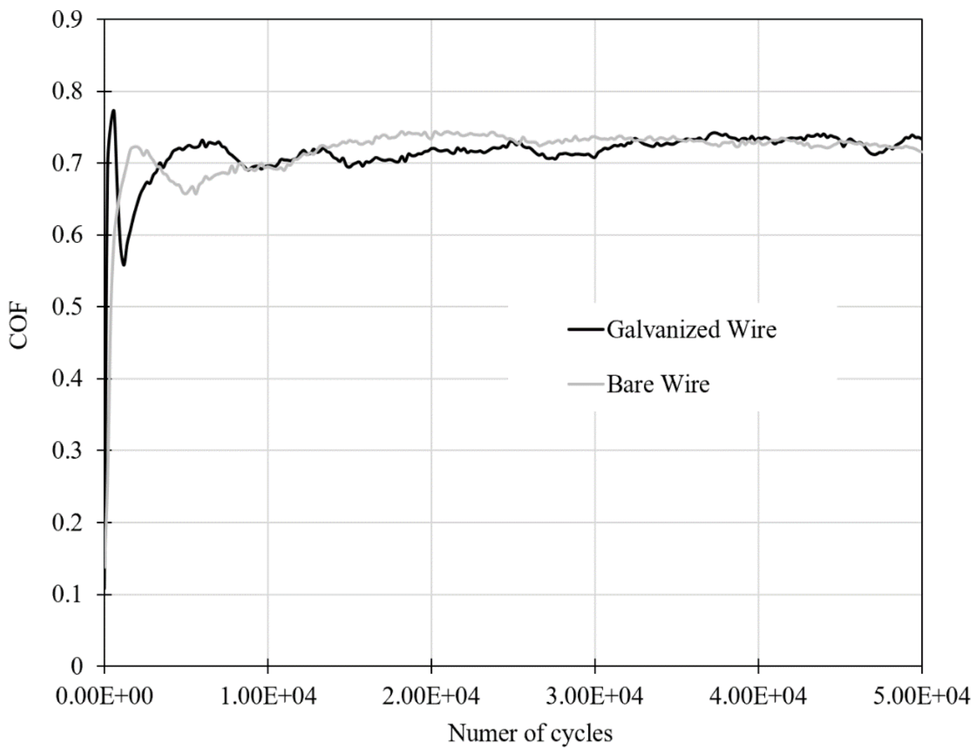


Figure 5-11 Comparing COF in the gross sliding regime for bare and galvanized wires.

5.6 Frictional force

The frictional force during the fretting fatigue tests was determined based on the difference between the load cell measurement at the top and the load cell measurement at the bottom. Figure 5-12 shows the frictional force versus the number of cycles in fretting fatigue tests in the stick-slip regime. Looking at the results, it can be seen that the frictional force is stable during the tests and no significant change can be seen in the amount of frictional force during the tests. Also, it can be seen that the frictional force significantly changes with a change in the load range (or remote stress range). However, a considerable difference cannot be seen between the results for different contact forces. This can be explained by the fact that, in the stick-slip regime, the axial force is lower than the threshold load that is required to completely move the wire along the pad. Therefore, increasing the contact force doesn't affect the frictional force. However, increasing the axial load (or remote stress range) does increase the calculated frictional force as F_{axial} is increasing.

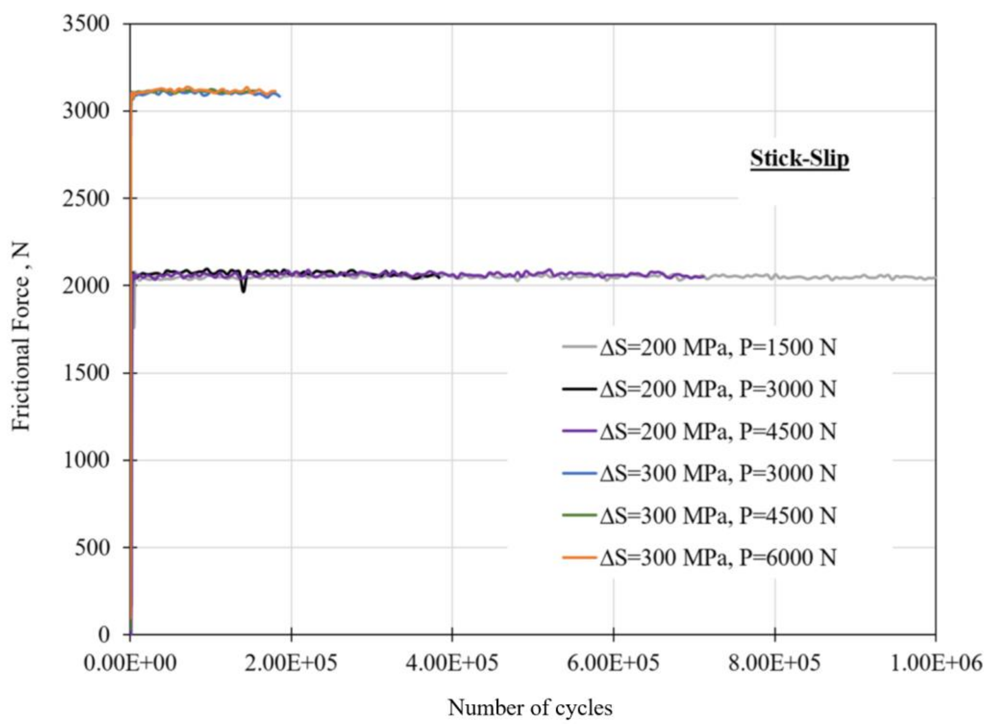


Figure 5-12 Frictional force measurements in the stick-slip regime, galvanized wires.

A different trend was observed for the tests in the gross sliding regime, the axial force is over the threshold load and the sliding occurs over the entire contact surface. Therefore, the frictional force increases with an increase in the contact force ($F_{frictional} = COF \times P < F_{axial}$). Looking at Figure 5-13, it can be seen that the frictional force increases with an increase in the contact force. Another observation in the tests in the gross sliding regime is that the frictional force decreases during the test. With a very high volume of wear in the tests in this regime, a considerable portion of the contact force transfers to the flexible side plates (shown in Figure 5-3), and as the wear increases, the contact forces decrease, and consequently, the frictional force decreases. This decrease in contact force has been previously seen in other fretting fatigue tests in the gross sliding regime (McColl et al. 2004).

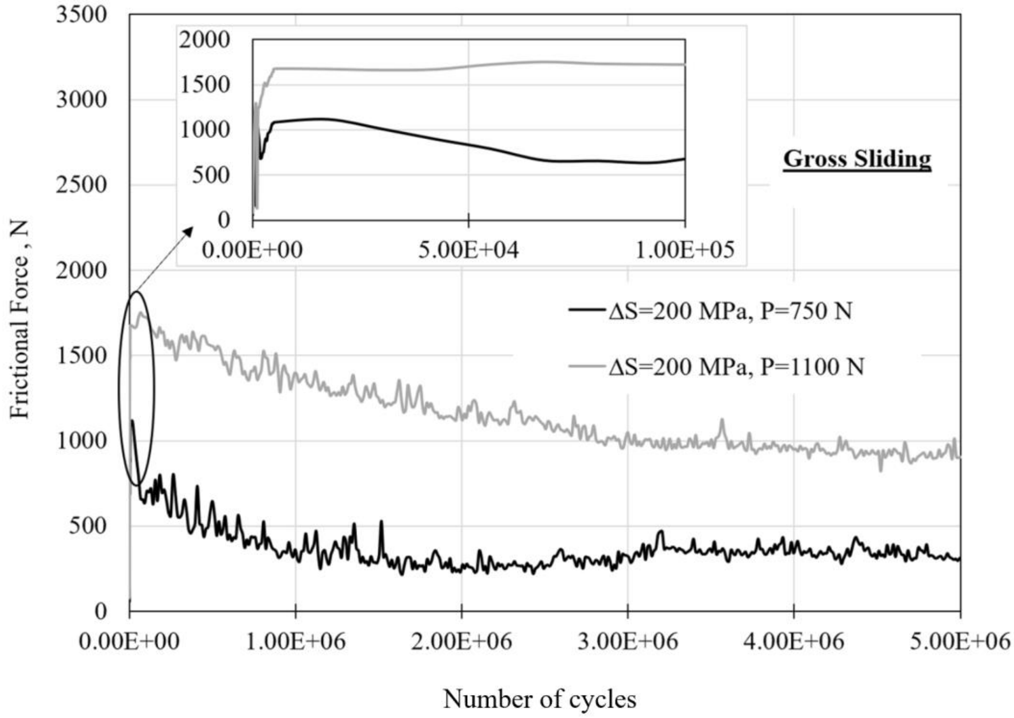


Figure 5-13 Frictional force for tests in the gross sliding regime, galvanized wire.

To exactly evaluate the flexibility of the side plates and the amount of force they carried during the tests, a dial gauge was used to determine the horizontal displacement of the side plates versus the applied load with no wire in the test frame. The load was recorded using the washer load cells on the threaded rods and a dial gauge was mounted to measure the horizontal displacement of the pads (see Figure 5-14 (a)). Also, a simple SAP 2000 frame model of the fretting apparatus, shown in Figure 5-14 (b), was used to evaluate the load vs. displacement curve. The flexible side plates and the holding box were modelled in the SAP 2000 frame model. The exact dimension of all test setup parts can be seen in Appendix B. Figure 5-15 shows the force vs. displacement graphs. It can be seen that the results of the frame mode are very close to the measured values using a dial gauge. Measurements were done from both sides of the test setup every couple of months during the testing period and no difference was seen between the results. The dial gauge was later used for the tests in the stick-slip regime. It was observed that the deformation of the plates due to plastic deformation and wear during the tests was very small (typically lower than 0.04 mm). Looking at Figure 5-15, it can be seen that the plates carry around 20-40 N of the contact force with this amount of deformation. This amount of force was less than 2.5% of the total contact force and it was considered in all of the tests at this regime during loading.

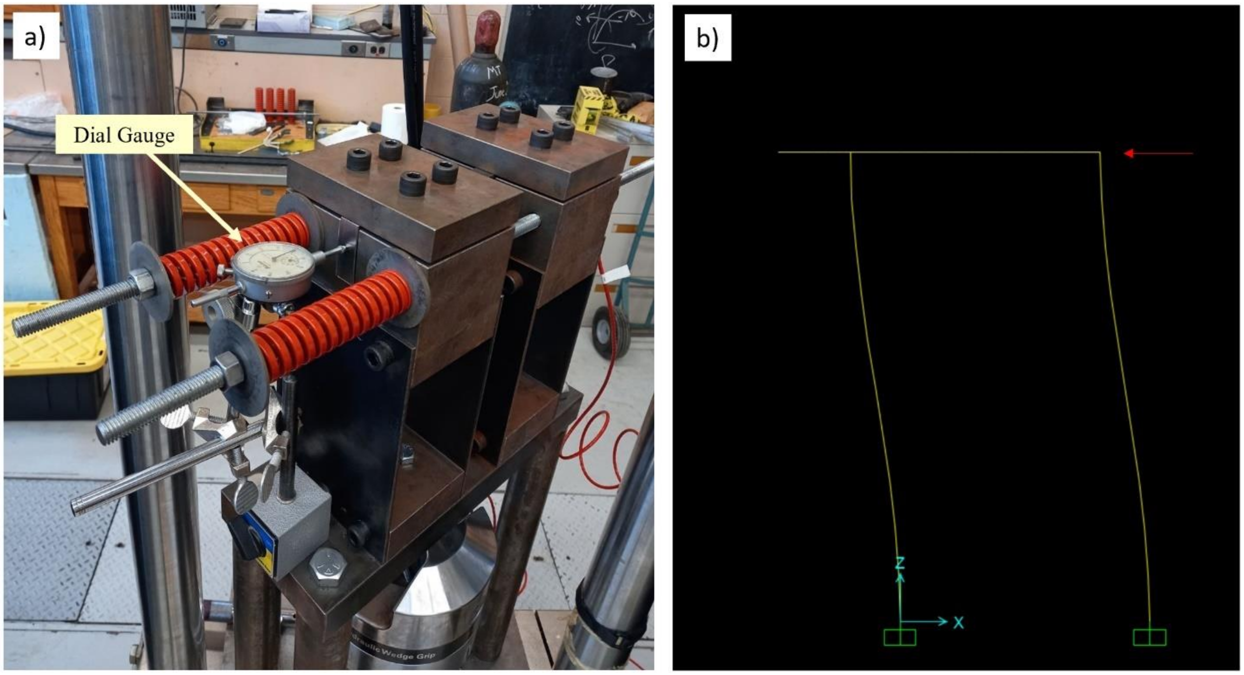


Figure 5-14 Dial gauge measurements of the horizontal deflection of fretting setup (a), SAP 2000 frame model of the setup.

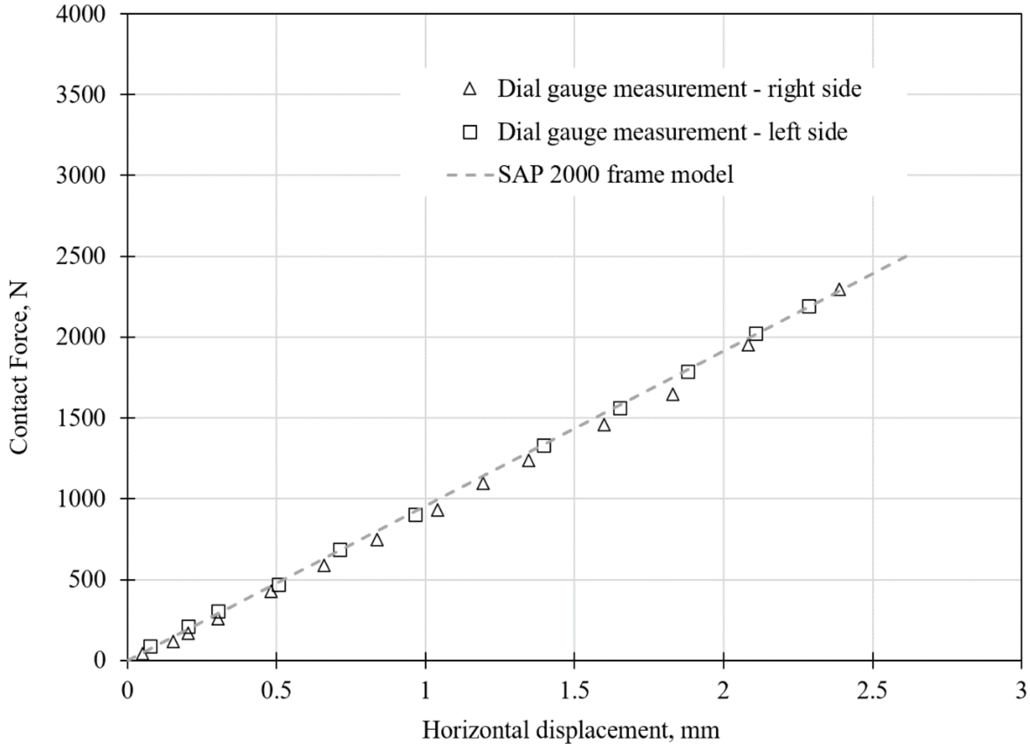


Figure 5-15 Horizontal flexibility of the test setup.

5.7 Slip displacement

Slip displacement is a critical parameter affecting the fretting fatigue life of the contacting components. This parameter has been used by Vingsbo and Söderberg (1988) to classify different fretting fatigue regimes. Also, it can be used in the numerical analysis of fretting fatigue problems. Therefore, this parameter is typically measured and reported with test results. In this work, the strain of the top part of the wire and the vertical flexibility of the setup was used to determine the slip displacement at the contact point (see Figure 5-16). The slip displacement can be determined as follows:

$$Slip\ Displacement = \frac{\Delta F_{top} \cdot L}{EA} - \Delta(F_f) \qquad \text{5-3}$$

where ΔF_{top} is the load range of the top part of the wire, E is the elastic modulus, A is the cross-section area of the wire, L is the length of the top part of the wire, and $\Delta(F_f)$ is the displacement of the pads due to the flexibility of the setup and can be measured based on the frictional force and the vertical flexibility of the setup. The force vs. displacement curve in the vertical direction of the test setup is shown in Figure 5-17. This curve was determined based on the SAP 2000 frame model shown in Figure 5-14 (b).

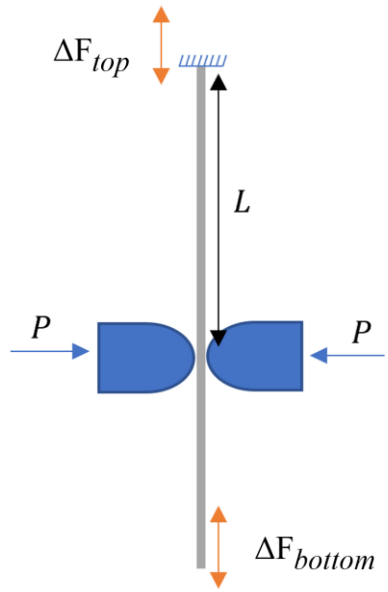


Figure 5-16 Schematic view of the wire in fretting fatigue tests.

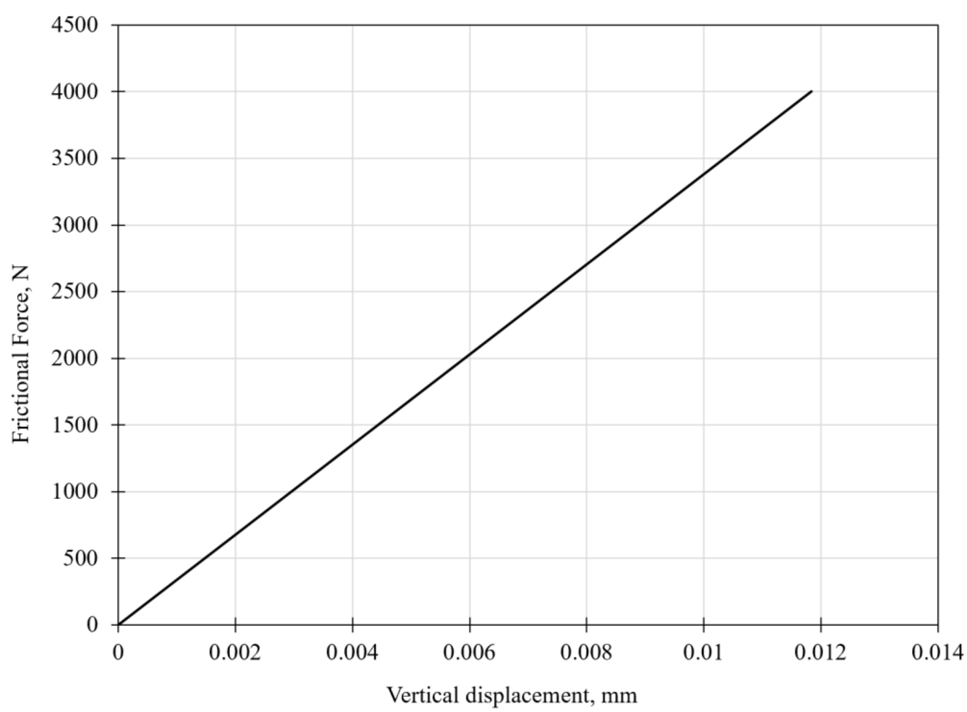


Figure 5-17 Force vs vertical displacement of the test setup using SAP 2000 frame model.

Similar to frictional force results, slip displacements were very consistent during the tests in the stick-slip regime. Figure 5-18 shows the slip displacement results for the tests in the stick-slip regime. Looking at these results, it can be seen that the slip displacement does not considerably change with the change in contact force. But it greatly depends on the remote stress range. As discussed in the frictional force section, in the stick-slip regime, changing the contact force does not considerably change the frictional force. Therefore, at the same stress range, all the effective parameters are very close to each other. However, increasing the remote stress range increases the tress range at the top part of the wire and consequently increases the slip displacement. Figure 5-19 shows the results for the gross sliding regime. As discussed before, because of the high volume of wear, the contact force and consequently the frictional force constantly decreases during these tests. Therefore, the load range at the top part of the wire increases and consequently the slip displacement increases during the tests.

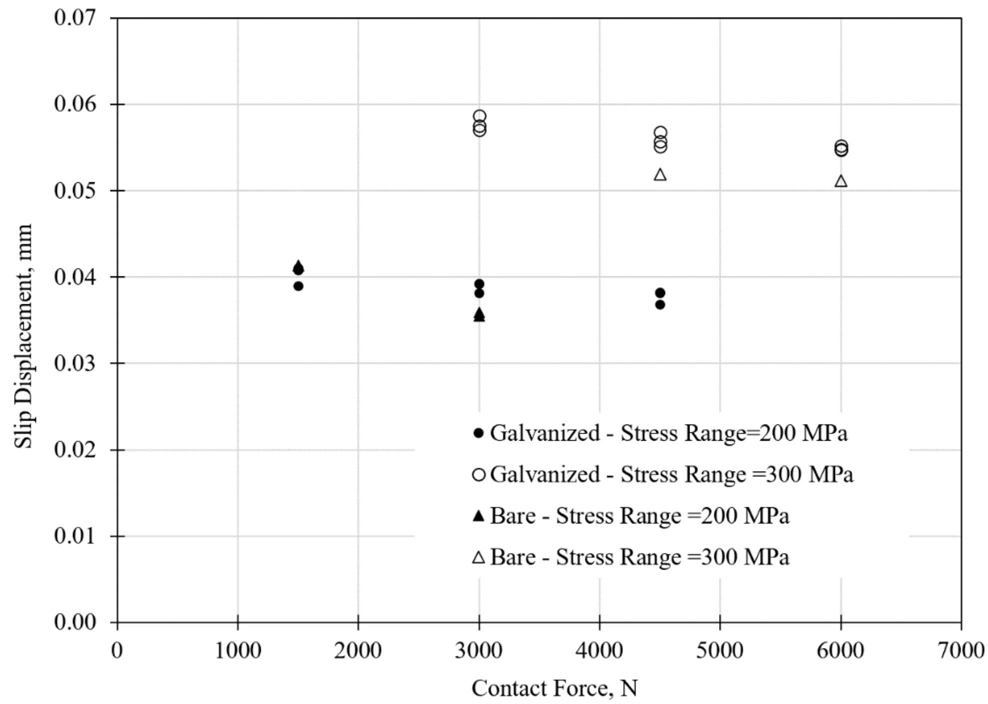


Figure 5-18 Slip displacement results in the stick-slip regime.

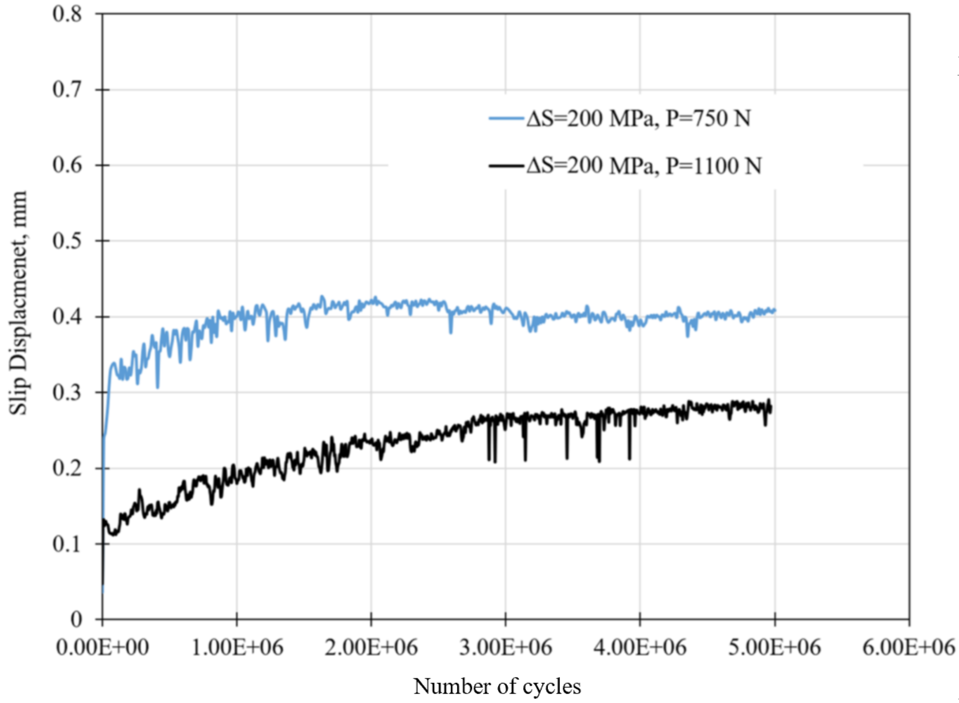


Figure 5-19 Slip displacement versus the number of cycles in the gross sliding regime, galvanized wires.

5.8 Material properties

As discussed in the previous section, there is a considerable difference between the fretting fatigue performance of the studied wires. One possible explanation was that the material properties of the wires may be different. Therefore, small hourglass samples from the central wire of cables were used to perform tensile/plain fatigue tests and check whether the cores of the wires have the same properties. Figure 5-20 shows the dimensions of hourglass samples of bare and galvanized wire.

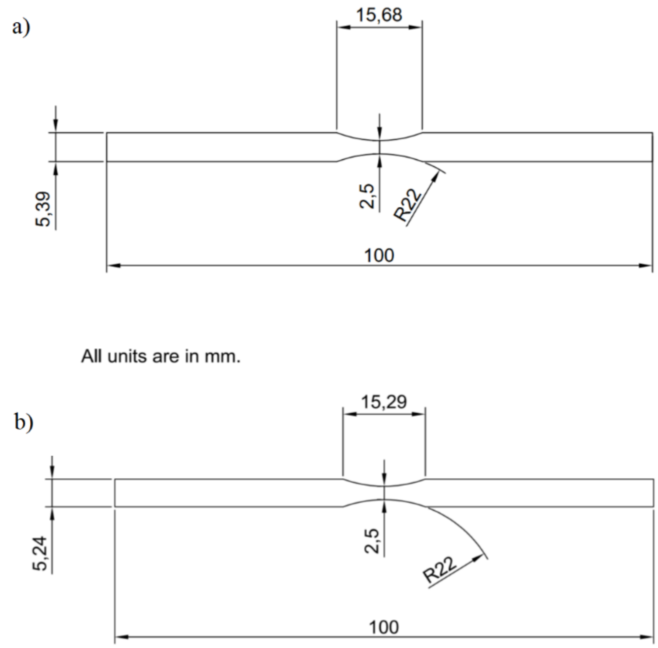


Figure 5-20 Drawing of hourglass samples of galvanized (a) and bare (b) wires.

The samples were first machined. However, the machined surfaces were not smooth. Roughness and small cracks can significantly affect the fatigue life of these specimens. To remove the machine marks, first, the samples were polished with #200 and #400 sandpaper. Then to get a mirror-like surface and remove small cracks, the samples were polished using a longitudinal polisher. Finally, all the samples were checked under a microscope to make sure that the surfaces of the samples were smooth (see Figure 5-21)

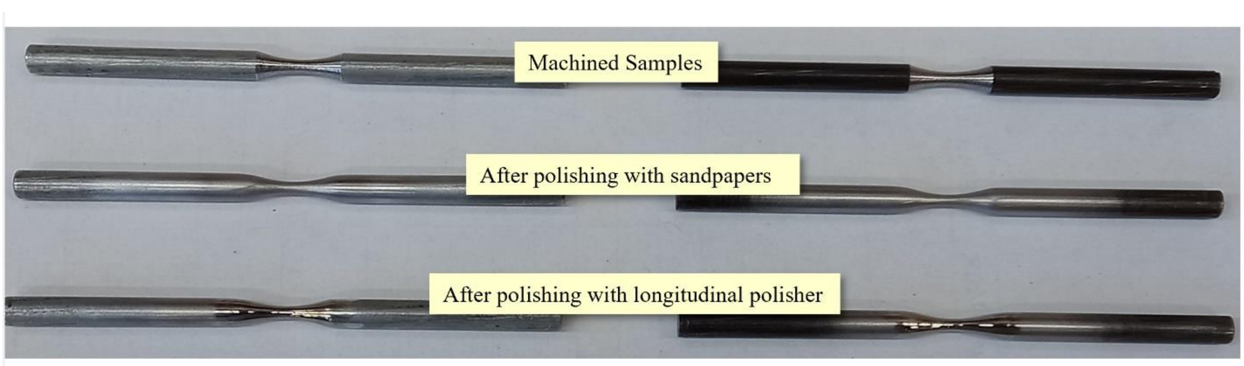


Figure 5-21 Hourglass sample preparation.

First, tensile tests were done on both wires. Figure 5-22 shows the tensile stress-strain curve of the bare and galvanized wire, respectively. The yield strength of both wires is very close. However, the galvanized wire is more ductile and has a higher fracture strain. The tensile material properties of the wires are listed in Table 5-2.

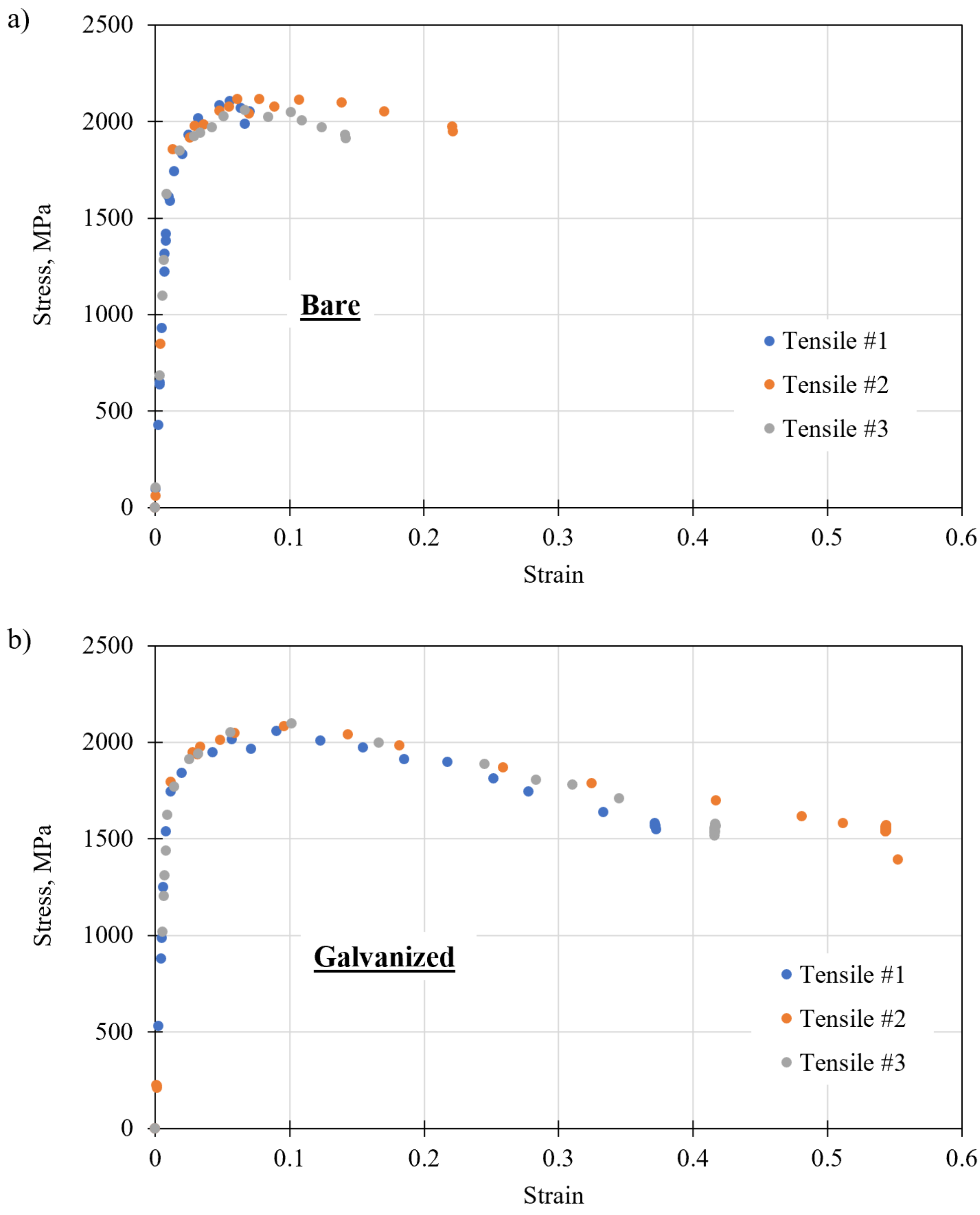


Figure 5-22 Tensile stress-strain curve of bare (a) and galvanized (b) wires.

Table 5-2 Material properties of bare and galvanized wires.

Specimen	Bare	Galvanized
Elastic modulus (MPa)	207044	195351
0.2% offset yield strength, S_y (MPa)	1756	1721
Ultimate tensile strength, S_u (MPa)	2147	2096
True fracture Strain, ε_f	0.128	0.395
Final engineering stress (MPa)	1997	1604
Fracture stress (MPa)	2214	2284
Reduction in Area (%)	9.71	29.29

Figure 5-23 shows the fatigue life data for both wire types. First, no considerable difference can be seen between the performance of the wire in the short-life region. However, interestingly, the fatigue limit of the galvanized wire is slightly higher than the bare wire. While the bare wire had a better performance in fretting fatigue tests, the material of the core of the galvanized wire has a higher fatigue limit. Table 5-3 summarizes the Coffin-Manson parameters of the wires. Figure 5-24 shows the typical failure surface of bare and galvanized wires. In the long-life region, the bare wires typically failed with crack initiation from the inside (fish-eye failure), however, the cracks typically initiated at the surface of galvanized samples.

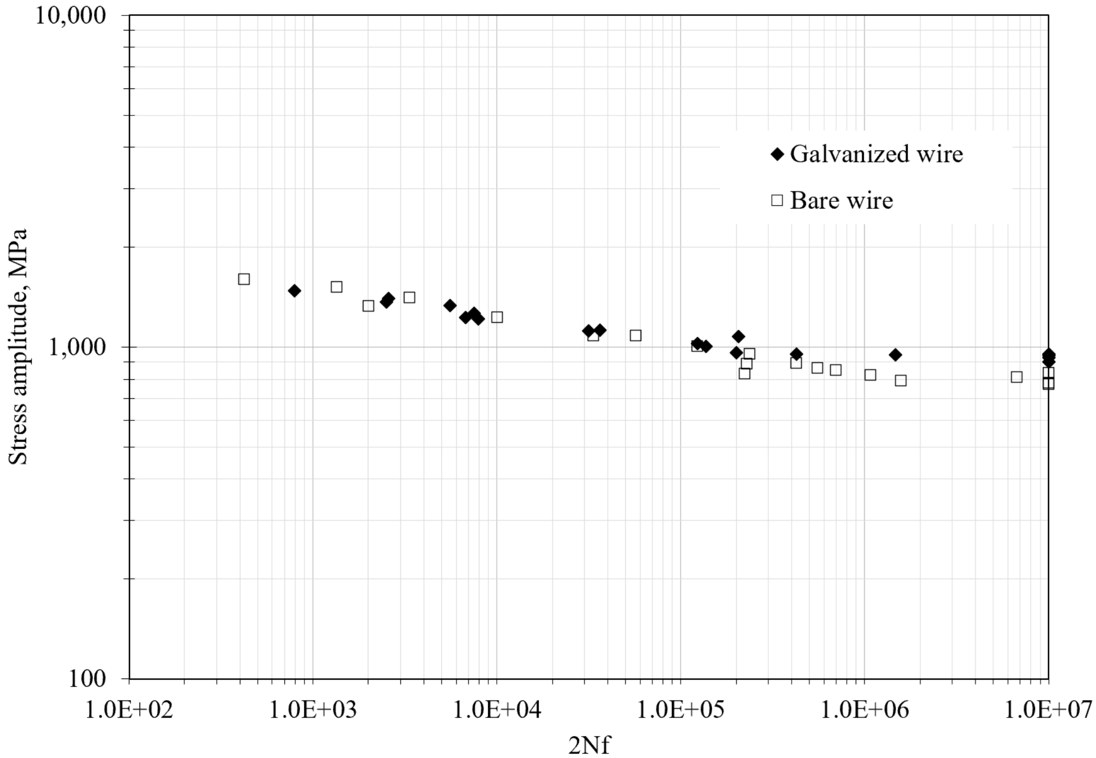


Figure 5-23 Stress life data for bare and galvanized wires.

Table 5-3 Coffin-Manson parameters for bare and galvanized wires.

Specimen	Bare	Galvanized
Fatigue strength coefficient, σ_f' (Mpa)	2675	2183
Fatigue strength exponent, b	-0.0859	-0.0657
Fatigue ductility coefficient, ε_f'	0.2067	1.99
Fatigue ductility exponent, c	-0.5047	-0.8092

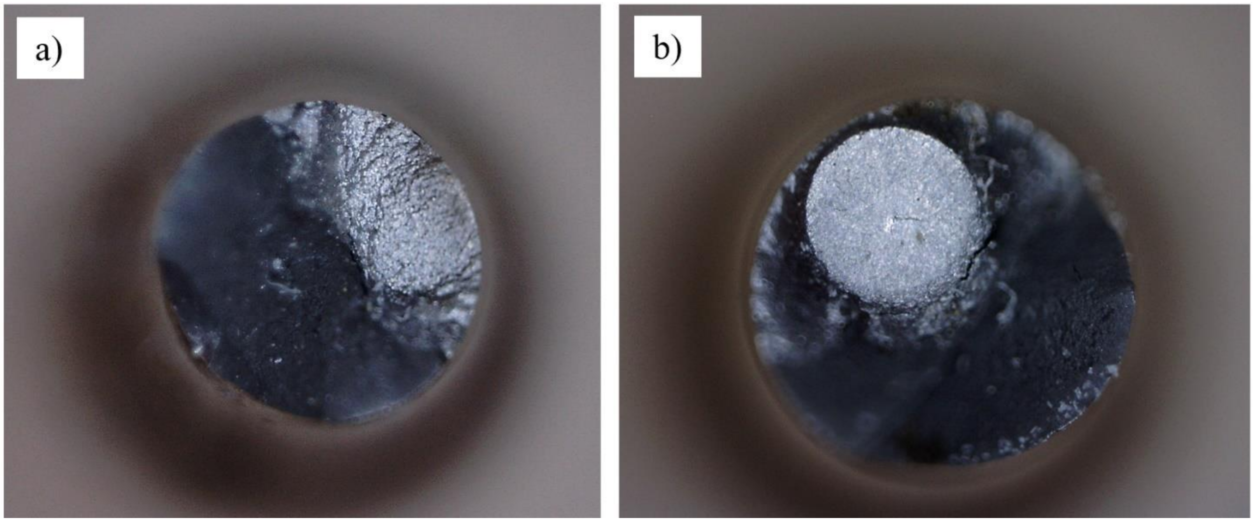


Figure 5-24 Typical fracture surface in galvanized (a) and bare (b) wires in the long-life region.

While the studies with hourglass samples provided valuable information about the tensile and fatigue properties of the wires, they still cannot explain the significant difference between the fretting fatigue performance of the wires. Therefore, it can only hypothesized there might be irregularities/defects at the surface of the galvanized wires that cause this difference.

5.9 Microstructure analysis

As discussed in the previous section, the study with hourglass samples did not explain the difference between the fretting fatigue performance of wires. Therefore, an evaluation of the surface structure of both wires was performed. In several previous works in the literature, it has been shown that galvanization can have a negative effect on fatigue performance of steel, specialty high-strength steel. It was explained mainly by the possible defects caused during the galvanization including brittle cracks in the galvanization layers or zinc penetration into the grain boundaries of steel (Bergengren and Melander 1992, Vogt et al. 2001, Sirin 2019). Given these defects, the cracks can propagate more easily at the surface of galvanized high-strength steel components. It should be noted that these studies evaluated the performance of high-strength steel components in air. In corrosive environments, however, galvanization has been proven to be a very effective protective layer. In this section, different tools and techniques were used to detect possible defects or differences at the surface of the studied wires.

5.9.1 Microhardness of the wire

Microhardness tests were performed to evaluate possible differences between the hardness of the studied wires. The microhardness tests were done using a 500 g load and 10 s dwell time. Figure 5-25 shows the microhardness test results of both wires. The tests were performed for a series of points with different distances to the surface. The average microhardness of bare and galvanized wires was 578 and 557 HV respectively. Microhardness results show that the bare wire is slightly harder than the galvanized wire. This difference in the microhardness of the wires can partially explain the difference between the fretting fatigue performance of the wires. The bare wire is harder. Therefore, crack initiation life is longer for the bare wire. With this difference in microhardness results, microscopic imaging was undertaken to get a better understanding of the surface microstructure of the two wire types.

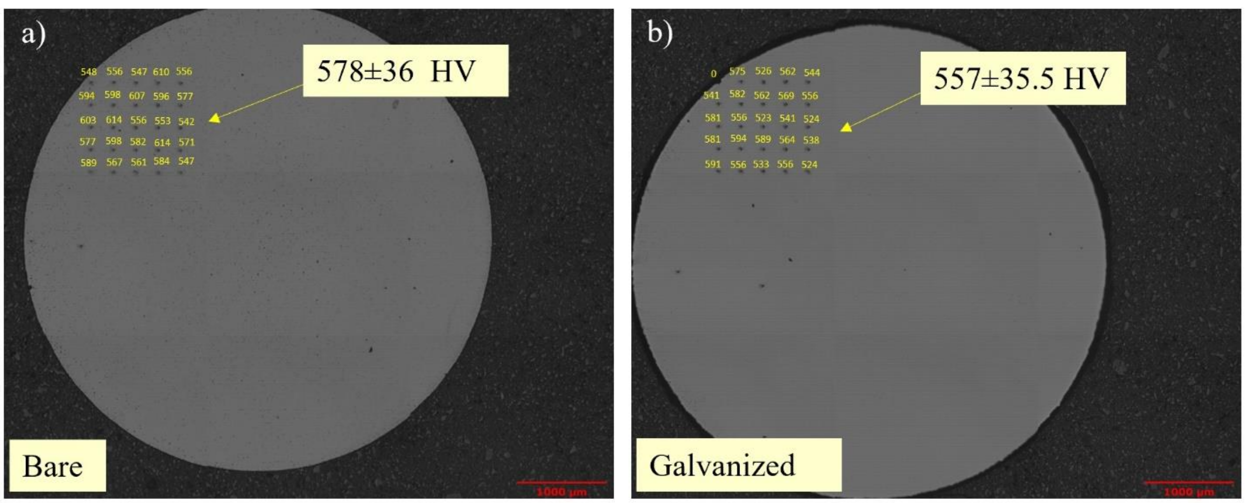


Figure 5-25 Microhardness tests results of bare and galvanized wires.

5.9.2 Surface profiles of the wires

Figure 5-26 shows surface profiles of the wires. Several studies showed that fatigue life decreases with an increase in surface roughness. Looking at these figures, it can be seen that the galvanized wire has a relatively rougher surface in comparison with the bare wire. This can be another explanation for the lower fatigue life of the galvanized wire, as there is a well-known direct relationship between higher surface roughness and shorter fatigue life.

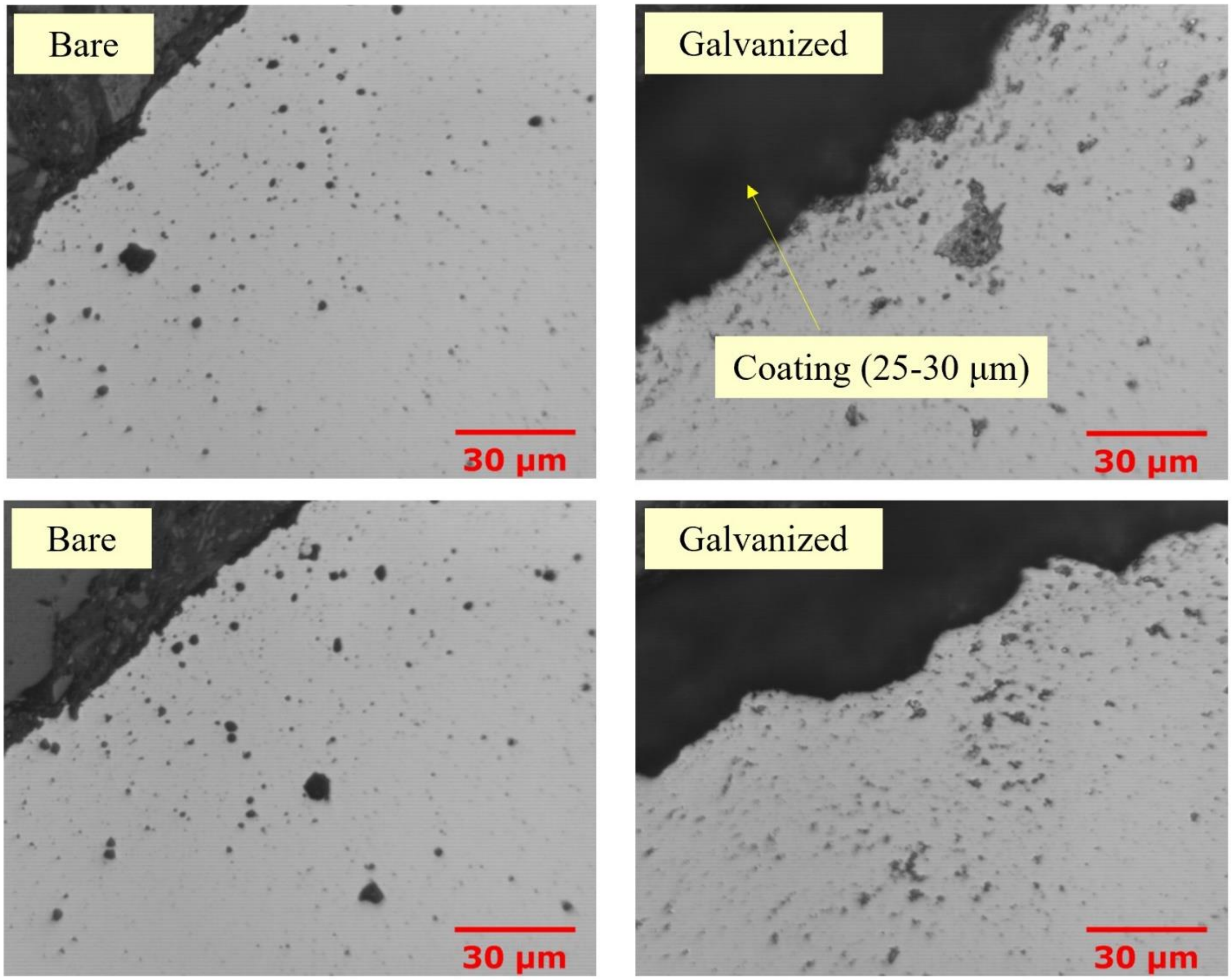


Figure 5-26 Comparing surface roughness of bare and galvanized wires.

5.9.3 SEM photos of the microstructure of the wires

Looking at the surface of the wires under a microscope was the next step in the microstructure analysis. SEM, scanning electron microscopy, photos of the surface of both wires can be seen in Figure 5-27. The bare wire has a uniform strong pearlite structure. However, looking at the surface of the galvanized wire, it can be seen that the pearlite structure is broken. Carbide and soft layers of ferrite can be seen in the photos. These soft layers of ferrite and broken carbide structure can be a suitable location for crack initiation. Given the high-stress concentration at the surface of wire in the fretting fatigue tests, the cracks can initiate much easier in these irregular structures in comparison with the uniform pearlite structure of bare wire. Also, in the fretting fatigue problem, there is a competition between crack propagation and wear. In galvanized wires, the cracks can freely grow in these irregular structures, while the wear is only affecting the coating. While, in the bare wire, the wear can remove small cracks that are initiating at the surface of the wire. However, in the stick-slip regime, the wear amount was very limited and considering the beneficial effect of wear requires detailed numerical analysis that includes modelling of wear. All in all, the SEM photos showed significant differences between the surface microstructure of both wires; the defects and irregularities at the surface of the galvanized wire can explain the considerable difference between the fretting fatigue performance of both wires.

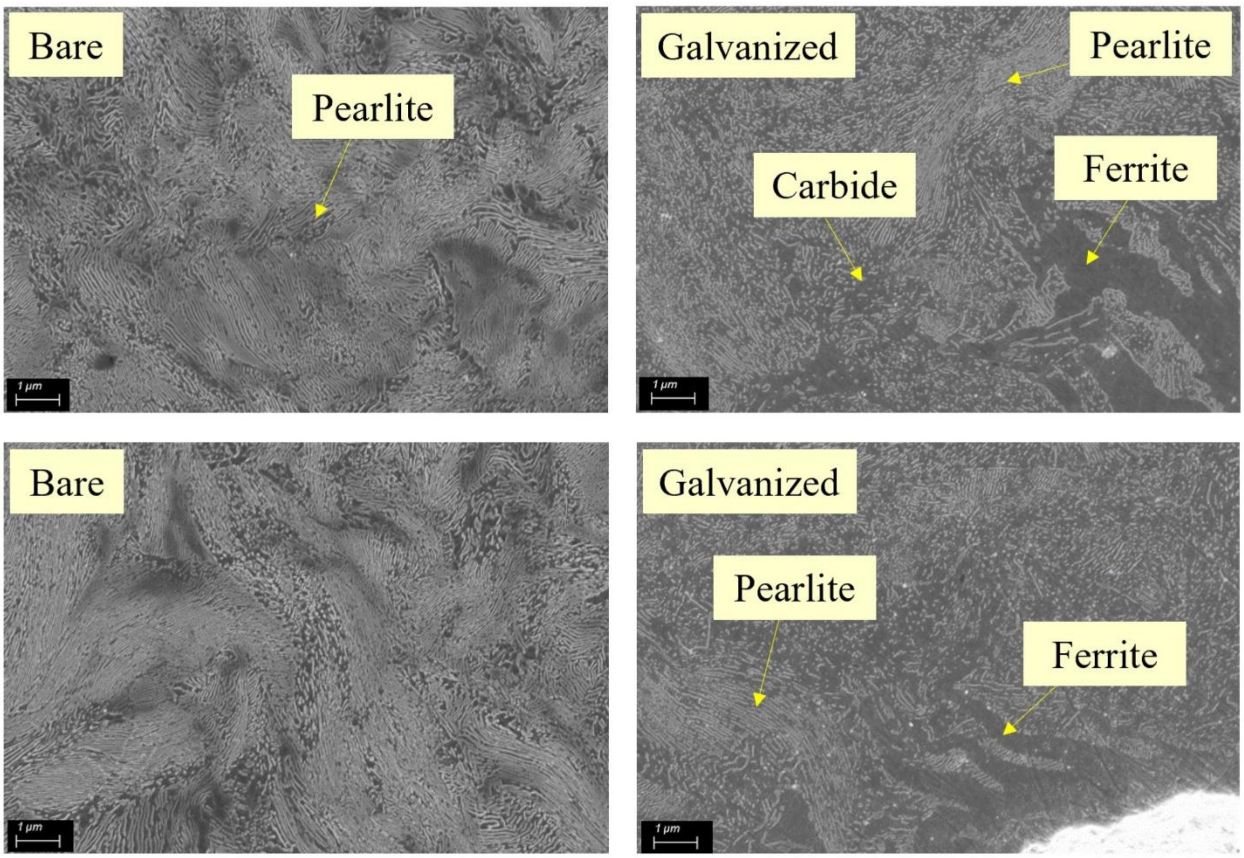


Figure 5-27 SEM photos of the bare and galvanized wires.

Another interesting observation in SEM photos was the pores in the bare wire. Looking at Figure 5-28, some pores can be seen in the bare wire. Plain fatigue results showed that the cracks typically initiate from inside of the bare wire rather than their surface. Also, the plain fatigue results showed that the bare wire has a slightly lower fatigue limit in comparison with the galvanized wire. These pores can be a good explanation for the lower fatigue limit of the bare wire.

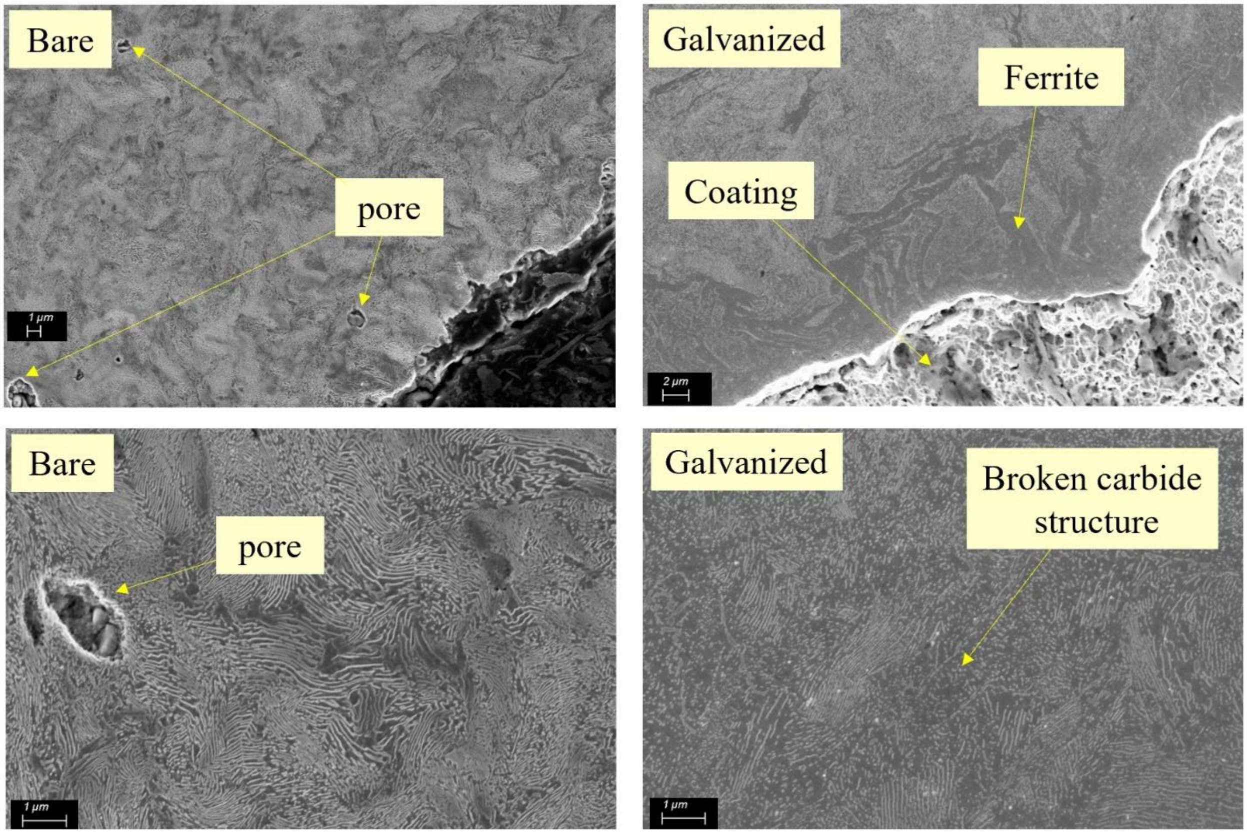


Figure 5-28 Defects in bare and galvanized wires.

All in all, the microstructure analysis showed several differences between the galvanized and bare wires. First, it was seen that the hardness of the galvanized wire was slightly lower than the bare wire Second, the galvanized wire had a relatively rougher surface in comparison with the bare wire. Third, the bare wire had a uniform pearlite structure at the surface. However, broken carbide structures and soft layers of ferrite were seen at the surface of galvanized wire. These differences can be a good explanation for a better performance of bare wire in fretting fatigue tests. Looking at the SEM photos, it also was seen that the bare wire has several pores inside them. These pores can explain the lower fatigue limit of the bare wire in comparison with the galvanized wire.

6. Analysis of Fretting Fatigue Tests Based on SWT Parameter

In this Chapter, the deterministic and probabilistic methods previously discussed in Chapters 3 and 4 were customized and applied to the fretting fatigue tests performed at the University of Waterloo using the small-scale fretting fatigue apparatus developed for the current thesis.

6.1 Deterministic fretting fatigue analysis of tests results

6.1.1 Fretting fatigue life determination based on SWT parameter

The analysis in this part is very similar to the one performed in Chapter 2 for saddle systems. Here, however, those methods are used to analyze the fretting fatigue tests performed at the University of Waterloo. A summary of the main parts is presented in the following paragraphs.

The SWT parameter was used to predict the fretting fatigue life of the tests presented in Chapter 5. The following equation was used to determine the fretting fatigue life based on the SWT parameter:

$$\mathrm{SWT} = \sigma_{\max} \cdot \Delta\varepsilon_a = \left(\sigma'_f\right)^2 \cdot \left(2 \cdot N_f\right)^{2 \cdot b} / E + \sigma'_f \cdot \varepsilon'_f \cdot \left(2 \cdot N_f\right)^{b+c} \qquad \text{6-1}$$

The following 3D stress/strain transformations were used to determine the critical plane (sum et al. 2005):

$$\sigma = \sigma_{11} \cdot n_x^2 + \sigma_{22} \cdot n_y^2 + \sigma_{33} \cdot n_z^2 + 2 \cdot \tau_{12} \cdot n_x \cdot n_y + 2 \cdot \tau_{23} \cdot n_y \cdot n_z + 2 \cdot \tau_{13} \cdot n_x \cdot n_z \qquad \text{6-2}$$

$$\varepsilon = \varepsilon_{11} \cdot n_x^2 + \varepsilon_{22} \cdot n_y^2 + \varepsilon_{33} \cdot n_z^2 + \gamma_{12} \cdot n_x \cdot n_y + \gamma_{23} \cdot n_y \cdot n_z + \gamma_{13} \cdot n_x \cdot n_z \qquad \text{6-3}$$

θ_h and θ_v are varied in 5° increments and $\sigma_{max} \cdot \Delta\varepsilon_a$ is calculated for each plane. The maximum value is identified and Equation 6-1 is then solved for N_f.

6.1.2 Finite element model

The FE program ABAQUS was employed to model the fretting fatigue tests. Figure 6-1 shows the model that was used in this study. One plane of symmetry was used to reduce the analysis time. The saddle/pad is curved to model a discrete contact point, the curvature of the pad type is equal to the curvature of an outer wire in a cable on a straight surface. Hard contact with the penalty algorithm was used for the normal contact behaviour. The tangential contact behaviour was controlled with the penalty method. Eight node linear brick elements (C3D8R) with an approximate size of 25 μm ×25 μm were used at the contact surface. An elastic material model with an elastic modulus of 195 Gpa and a possion's ratio of 0.3 was assumed for the galvanized wire. For bare wire analysis, an elastic material model with an elastic modulus of 207 Gpa and a possion's ratio of 0.3 was used. An elastic-fully plastic material model with an elastic modulus of 200 Gpa and yield stress of 500 MPa was used for the saddle/contacting pad. The inputs/boundary conditions of the model are slip displacement, contact force, remote stress, and coefficient of friction. These parameters were directly measured during the test and were used as the inputs for the studies in this part (see Table 6-1)

The loading was applied in four steps. According to the tests performed at the University of Waterloo, in the first step, the minimum normal stress is applied (537 or 637 MPa based on the test). However, to ensure contact and avoid convergence issues, a small amount of contact force is also applied in the first step. Then, the actual value of the contact force is applied in the second step. Following that, in the third step, the maximum normal stress (837 MPa) and the slip displacement were applied. The slip displacement was applied by moving S1 and S3 surfaces. In the last step, the normal stress reduces to its minimum value. Also, S1 and S3 surfaces return to their original position. The W3 surface was fixed in y and z directions and it only can be moved in the x direction at the top. The initial increment size in all steps was set to 0.001. The maximum step size was set to 0.1. the number of increments varies based on the convergence in increments. The direct equation solver with the full newton solution technique implemented in ABAQUS was used for the analysis. The loading and unloading stages were applied in two different steps to enable the recording of the stress/strains at the end of each stage. The results at the end of the third (loading) and fourth (unloading) steps were used to determine the SWT parameter.

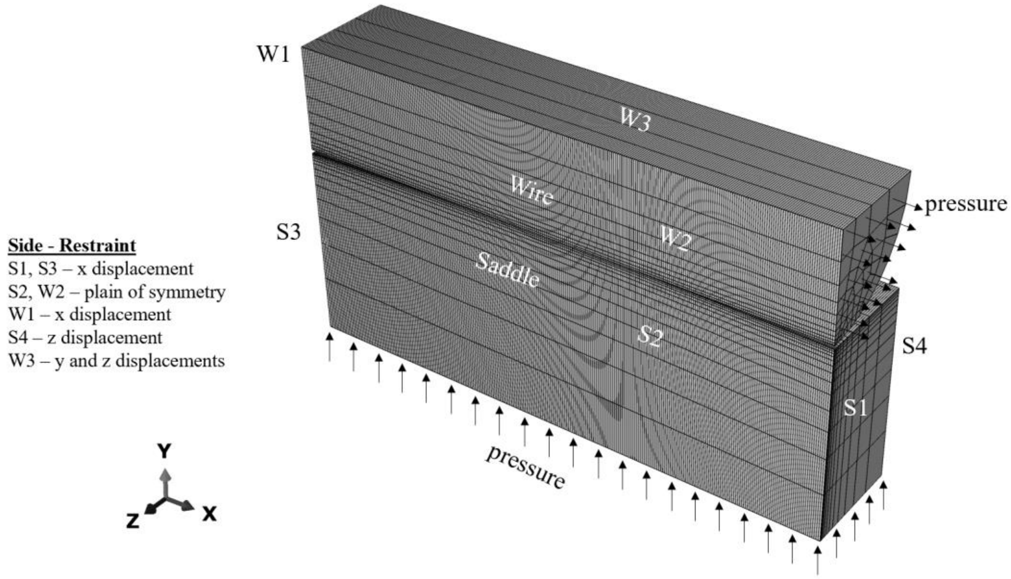


Figure 6-1 FE model of the contact point in fretting fatigue tests.

6.1.3 Coffin-Manson parameters

Coffin-Manson parameters for bare/galvanized wires were based on the materials tests presented in Chapter 5 and are listed in Table 6-1.

Table 6-1 Coffin-Manson properties of bare and galvanized wires.

Specimen	Galvanized	Bare
Fatigue strength coefficient, σ_f' (MPa)	2183	2675
Fatigue strength exponent, *b*	-0.0657	-0.0859
Fatigue ductility coefficient, ε_f'	1.99	0.2067
Fatigue ductility exponent, *c*	-0.8092	-0.5047

6.1.4 Coefficient of friction

As the COF in stick-slip is not determined, the analysis was performed with two COFs for each test: the gross sliding COF which is 0.75, and the quasi-COF, which is the ratio of the frictional force over the contact force. The actual average COF is somewhere between these two values, and these two COF are upper/lower bounds for the actual average COF.

6.1.5 Fretting fatigue life predictions based on the SWT parameter

Figure 6-2 shows the fretting fatigue life predictions based on the SWT parameter versus test results for the galvanized and bare wires. First, looking at the results for the galvanized wire, it can be seen that the predictions based on the SWT parameter and the COF of 0.75 are conservative and are a lower bound for the test results. Also, it can be seen that the results based on the quasi-COF are an upper bound for most of the tests. Looking at the results for the bare wire, similarly, it can be seen that the results based on the quasi-COF are higher than the results based on the COF of 0.75. However, they predict a finite fatigue life in the long-life domain for the bare wires, while the actual wires did not fail in these fretting fatigue tests.

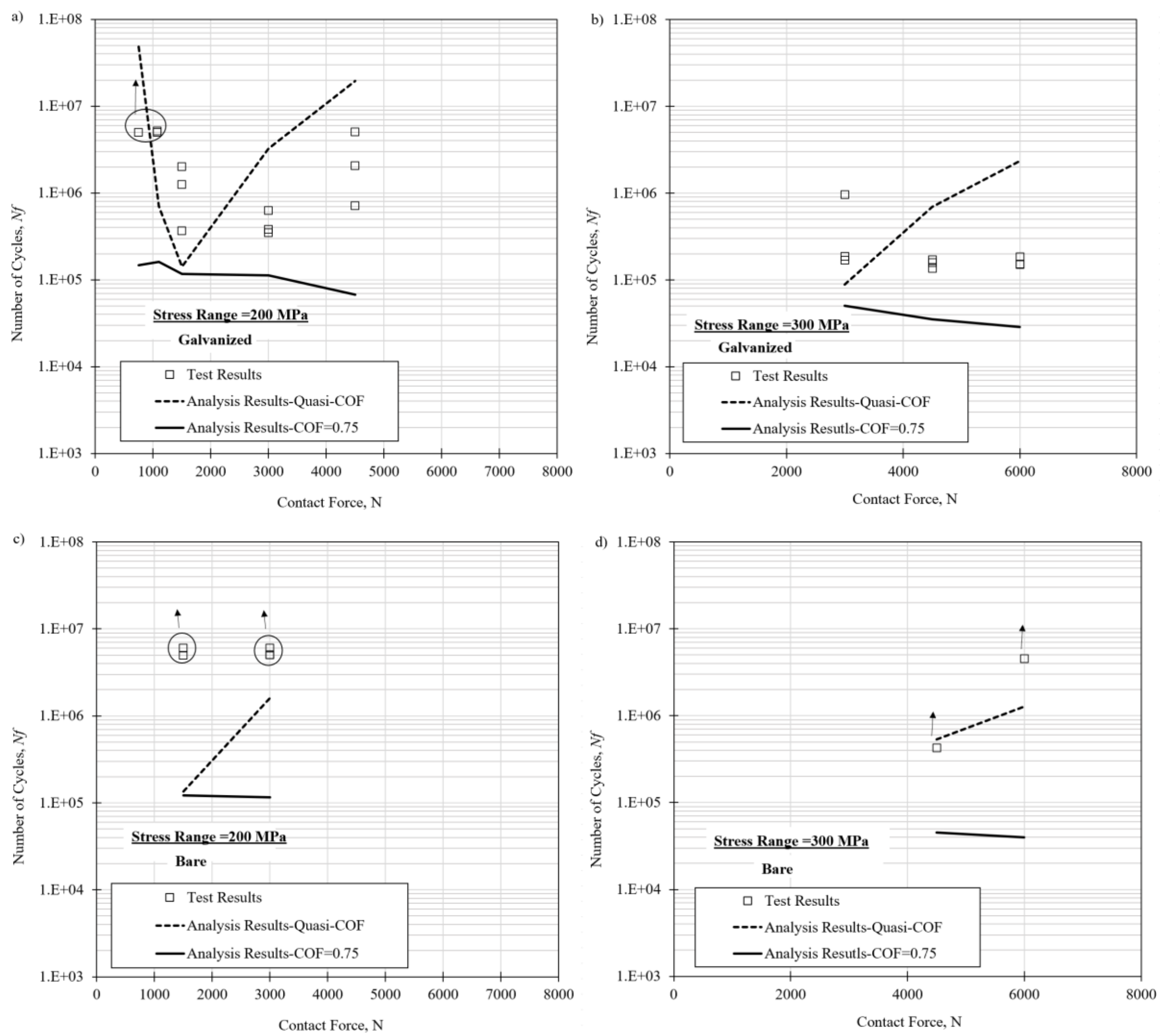


Figure 6-2 Fretting fatigue life predictions based on the SWT parameter for galvanized (a, b) and bare (c, d) wires.

As discussed before, the bare wire includes pores, which decrease their plain fatigue life. However, cracks initiate from the surface in fretting fatigue problems. Therefore, employing the SWT parameter based on the Coffin-Manson parameters obtained on tests of the wire core material does not lead to accurate results. This issue becomes more apparent when we compare the results for both wires. Figure 6-3 compares SWT vs. fretting fatigue life curves of bare/galvanized wires. It can be seen that at the same SWT in long-life domain, the fretting fatigue life predictions for the bare wire are lower than those for the galvanized wire. All in all, the following issues were found related to using the SWT parameter obtained from the core material for fretting fatigue life predictions of the tested wires in this study:

- The SWT parameter does not account for the presence of defects/cracks at the surface of the wire specimens. In this work, it was observed that the galvanized wire has defects at the surface, while the bare wire was uniform.
- Coffin-Manson parameters based on fatigue tests of hourglass shaped wire samples can be affected by internal pores/defects, while the fretting fatigue is more related to the properties at the surface of the wire. In this work, the bare wire had internal pores but a uniform structure at the surface.

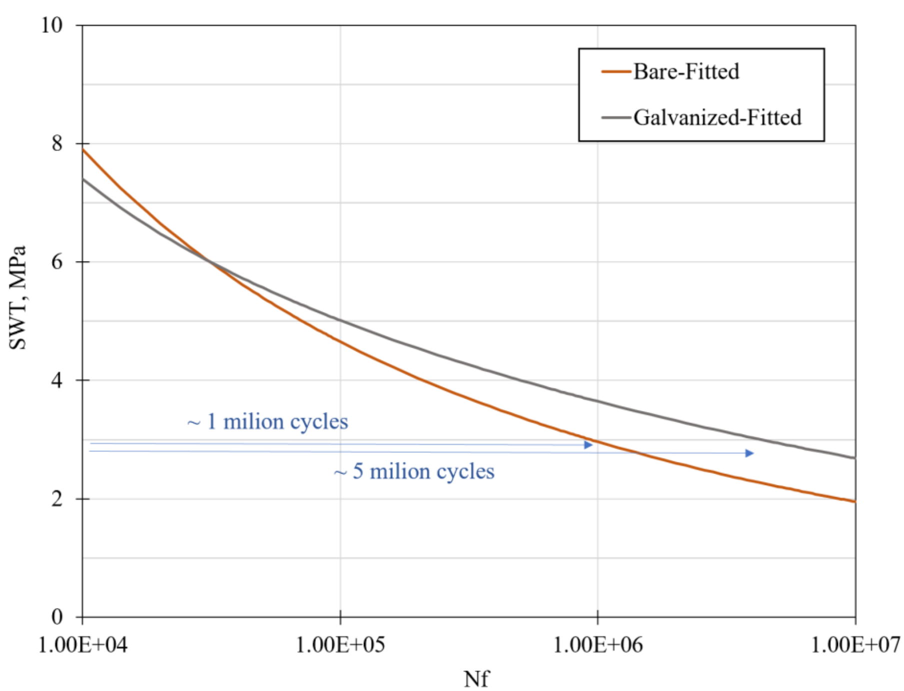


Figure 6-3 Sensitivity of fretting fatigue life to Coffin-Manson parameters based on bare/galvanized properties.

One possible approach to overcome the issue with the Coffin-Manson parameters based on the inside pores could be using the hardness/UTS of the wires to determine the Coffin-Manson properties based on the empirical equations in the literature. Empirical equations in the literature offer a way to approximate the Coffin-Manson parameters based on hardness. However, using this approach eliminates the issues related to the inside pores affecting the bare wire.

Given the issues found in determining Coffin-Manson parameters based on plain fatigue tests, it was decided to try the empirical equations suggested in the literature for determining the Coffin-Manson parameters based on the hardness, ultimate tensile strength, etc. These empirical equations were previously listed in Table 3-1. To compare these equations, it was decided to plot strain amplitude vs. the number of reversals based on different models and compare them with the fitted curves to the test results. In other words, the following equation was plotted using different sets of Coffin-Manson parameters based on empirical equations.

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma_f'}{E} \cdot \left(2 \cdot N_f\right)^b + \varepsilon_f' \cdot \left(2 \cdot N_f\right)^c \qquad \text{6-4}$$

The results are shown in Figure 6-4. It is expected that the results based on the empirical models will be close to the galvanized wire fitted curve. Looking at these curves, it seems the curve based on the parameters suggested by Meggiolaro and Castro (2004) is the closest, especially in the long-life region, which is of more interest in this study. To further evaluate these models, it was decided to plot the SWT parameter vs. the number of reversals. In other words, Equation 6-1 was plotted using the different sets of Coffin-Manson parameters.

Looking at the results in Figure 6-5, the same trend can be seen in the results – a good match between the curves presented by Meggiolaro and Castro (2004) and the fitted curve. Another observation from these figures is that all the models predict close results for bare and galvanized wires, especially in the long-life domain. Therefore, it can be argued that galvanized wire properties also can be a good approximation for the bare wire.

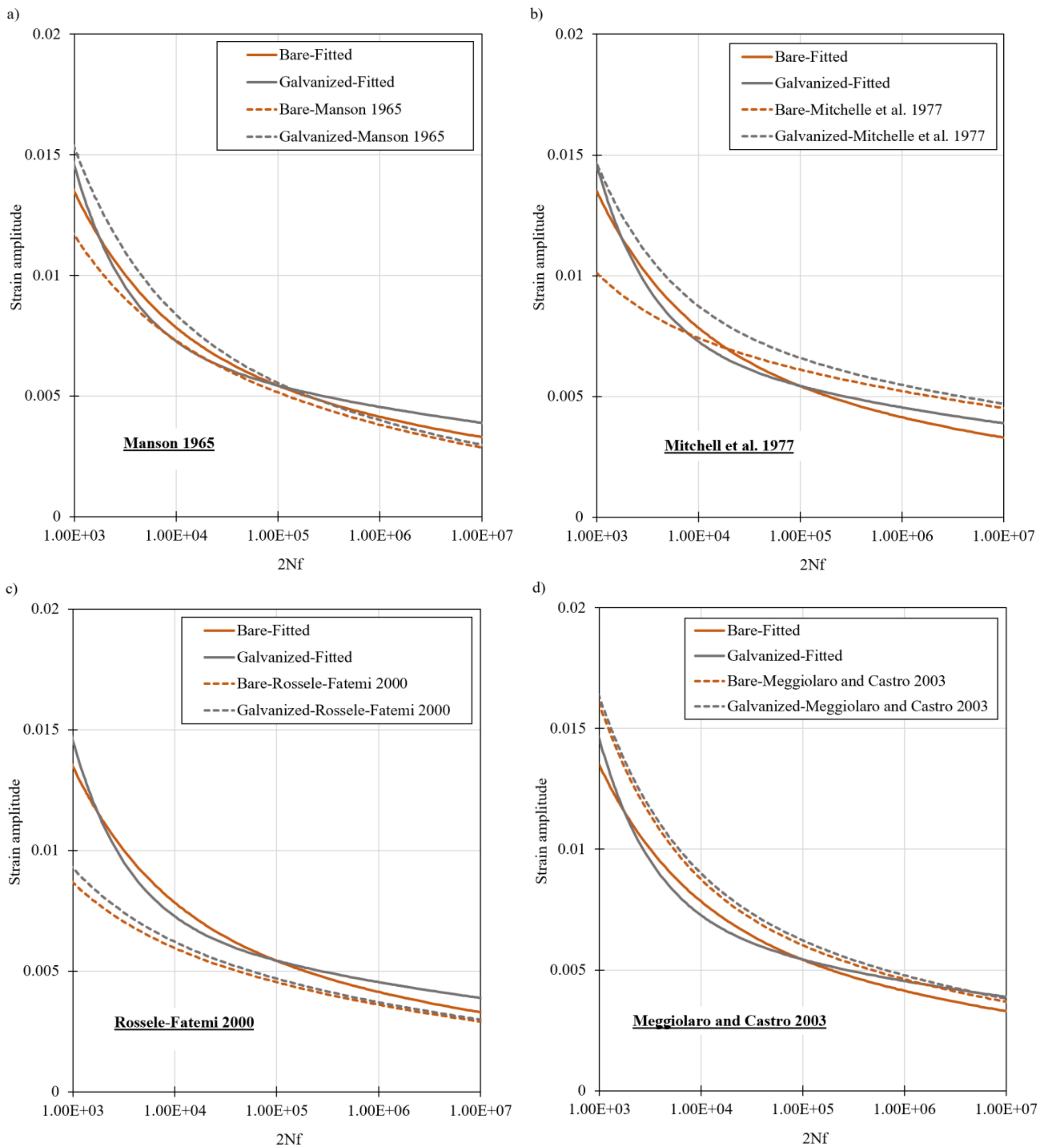


Figure 6-4 Strain vs. number of reversals based on different empirical models.

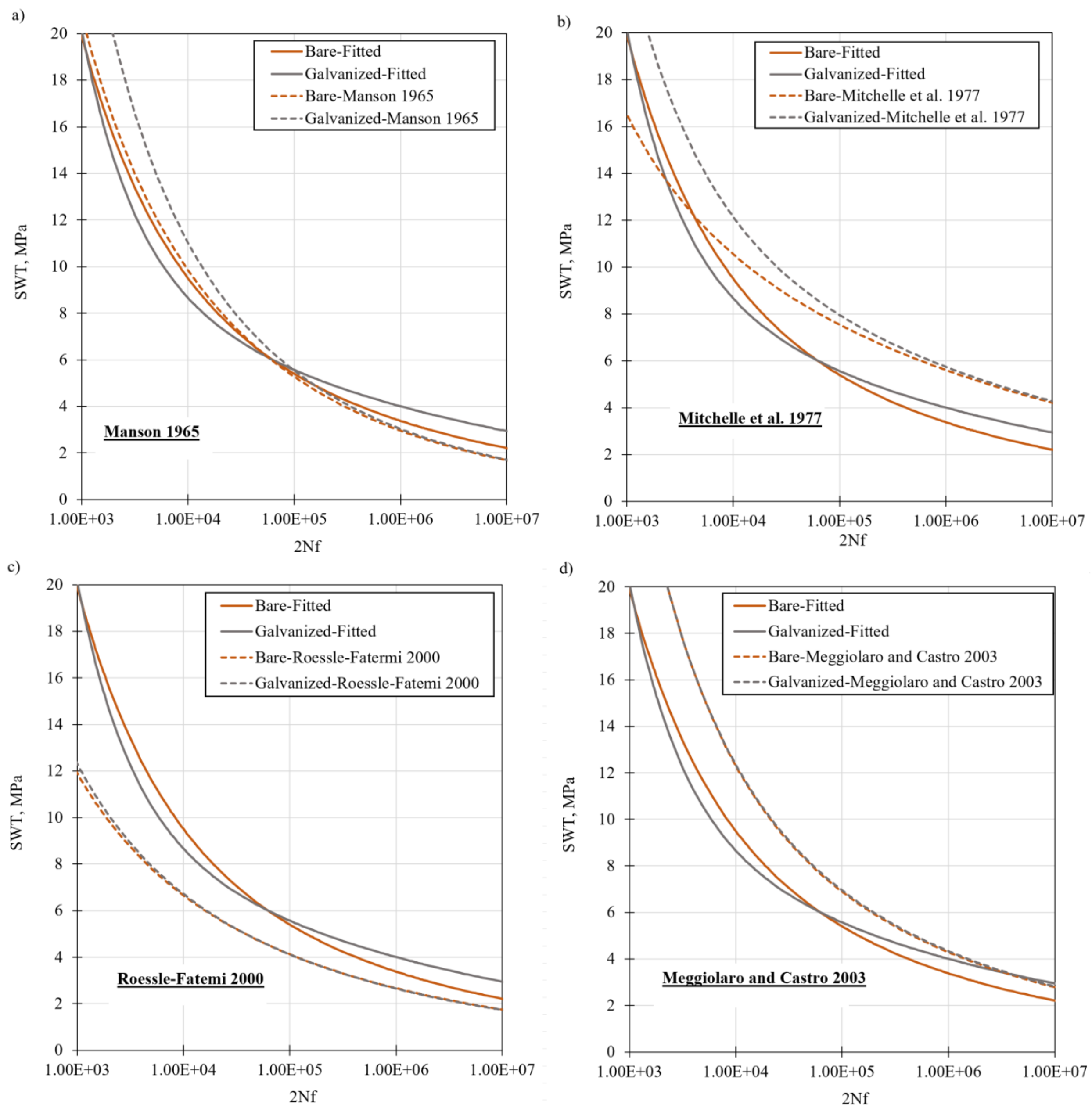


Figure 6-5 SWT vs. the number of reversals based on different empirical models.

Given the comparisons in Figure 6-4 and Figure 6-5, the analysis for the bare wire was repeated based on the SWT curve suggested by Meggiolaro and Castro (2004) and the SWT curve based on the galvanized wire properties. The results are summarized in Figure 6-6. It can be seen that higher fretting fatigue life is predicted based on the modified Coffin-Manson properties. It should be noted that these properties are still only approximations. The bare wire without pores might have a better performance. All in all, it seems that the issues with relating the Coffin-Manson properties of wires with defects to the fretting fatigue life prediction based on the SWT parameter can partially explain the lower fretting fatigue life predictions for the bare wire in comparison with tests. However, no bare wire failed; it seems that the main reason for the different performance of the galvanized and bare wires is due to the effect of surface properties. Also, all the fretting fatigue life predictions and test results of the galvanized and bare wires are in the long-life domain. In this domain, small differences between the wire properties (e.g., surface defects, differences in surface hardness, etc.) can easily shift the fatigue life of a specimen between a long life and a run-out.

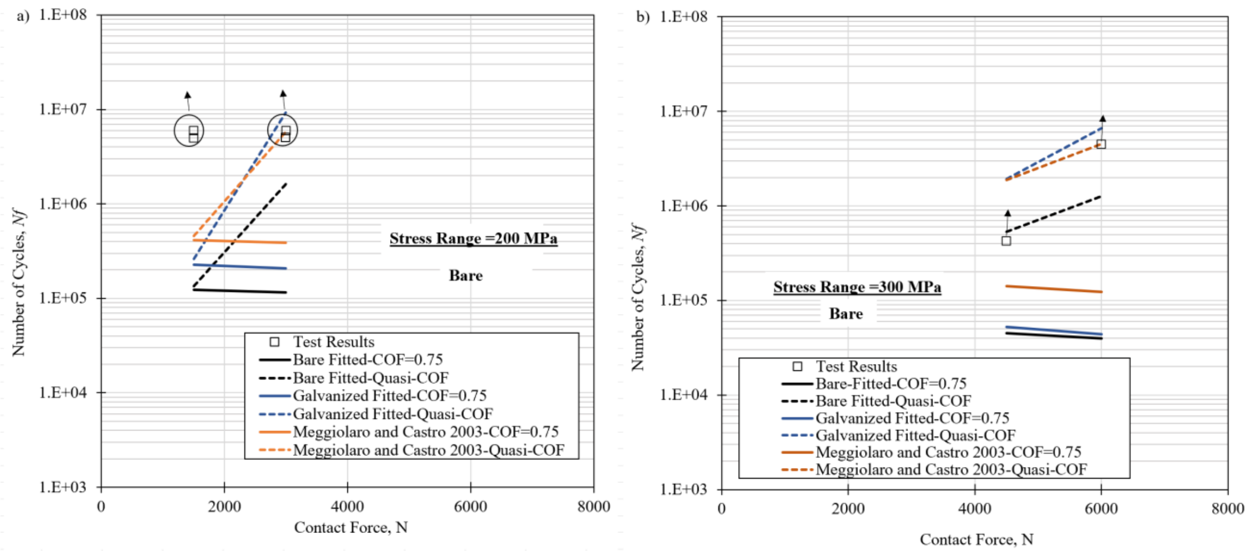


Figure 6-6 Fretting fatigue life predictions for bare wire based on different SWT-life curves.

6.2 Probabilistic analysis of the fretting fatigue tests

Following the deterministic evaluation of the problem, given the uncertainties in the material properties, coefficient of friction, and the scatter that was seen in the results of the test, it was decided to perform the same analysis, but using the probabilistic framework presented earlier to consider the uncertainty in the material properties and the COF. The analysis in this section is similar to the one presented in Chapter 4, however, given the fact that the contact force and slip displacement were measured during the tests, no bias factor was assumed for these two parameters. Similar to Chapter 4, the analysis was performed using MCS and MDRM.

6.2.1 Problem definition and objective

The main objective of this analysis was to determine the PDF and the CDF of fretting fatigue life, and then the survival probabilities associated with different applied load levels, for the fretting fatigue tests performed at the University of Waterloo. The fretting fatigue life, N_f was determined based on the SWT parameter using the following equation:

$$\text{SWT} = \sigma_{\max} \cdot \Delta\varepsilon_a = \left(\sigma'_f\right)^2 \cdot \left(2 \cdot N_f\right)^{2\cdot b} / E + \sigma'_f \cdot \varepsilon'_f \cdot \left(2 \cdot N_f\right)^{b+c} \qquad \text{6-5}$$

Looking at this equation, on the left side, the SWT is a parameter that is based on stresses and strains obtained from finite element analysis of the problem. On the right-hand side, N_f is the fretting fatigue life, E is the elastic modulus, and the rest of the variables are the Coffin-Manson parameters which are defined and discussed in the following paragraphs.

6.2.2 Analysis parameters

6.2.2.1 SWT parameter

The SWT parameter was determined based on the stress/strains from finite element analysis of the fretting fatigue tests. The finite element model used is discussed in detail in Section 6.1.2. The input parameters of this finite element model are based on the fretting fatigue tests and were measured during the tests. The main uncertainty was found to be in the COF. An upper and lower bound value of COF could be found based on the fatigue tests (i.e. the gross slip COF and the "quasi-COF" in the stick-slip domain). With no basis available to characterize the COF distribution more precisely, a uniform distribution was used for this parameter.

6.2.2.2 Variables related to material properties

Fatigue strength coefficient, σ'_f, fatigue strength exponent, b, fatigue ductility coefficient, ε'_f, and fatigue ductility exponent, c, are material properties related to the fatigue performance of materials. Based on Zhu et al. (2017), b and c were assumed to be deterministic and a lognormal distribution with a COV of 0.05 and 0.16 was assumed for σ'_f and ε'_f respectively. The average of the parameters was determined based on the fatigue tests in the current study using the two different bridge cable types: galvanized and bare.

The parameters of the problem are listed in Table 6-2.

Table 6-2 Statistics of parameters for probabilistic analysis.

Parameter	Wire Type	Distribution	Average	COV
Coefficient of friction, μ	Bare/ Galvanized	Uniform [COF_{quasi} -0.75]	-	-
Fatigue strength coefficient, σ_f' (MPa)	Bare	Lognormal	2675	0.05
Fatigue strength exponent, b	Bare	-	-0.0859	-
Fatigue ductility coefficient, ε_f'	Bare	Lognormal	0.2067	0.16
Fatigue ductility exponent, c	Bare	-	-0.5047	-
Fatigue strength coefficient, σ_f' (MPa)	Galvanized	Lognormal	2183	0.05
Fatigue strength exponent, b	Galvanized	-	-0.0657	-
Fatigue ductility coefficient, ε_f'	Galvanized	Lognormal	1.99	0.16
Fatigue ductility exponent, c	Galvanized	-	-0.8092	-

6.2.3 Analysis frameworks

Given the fact that each FE analysis takes a few hours to run, crude MCS is not practical. Therefore, a framework based on MDRM was employed in this study. Figure 6-7 shows how each variable plays a role in determining the fretting fatigue life of wire for tests performed at the University of Waterloo. It should be noted that this figure only shows the variable parameters. The deterministic parameters were measured during the tests and applied in the FE model. Also, other material properties including elastic modulus, fatigue strength exponent, and fatigue ductility exponent were assumed to be constant. Given that only one variable affected the FE model and SWT determination, only five FE analyses were required for each test. However, a total of 15 points, five for each variable, were used in the analysis for each test.

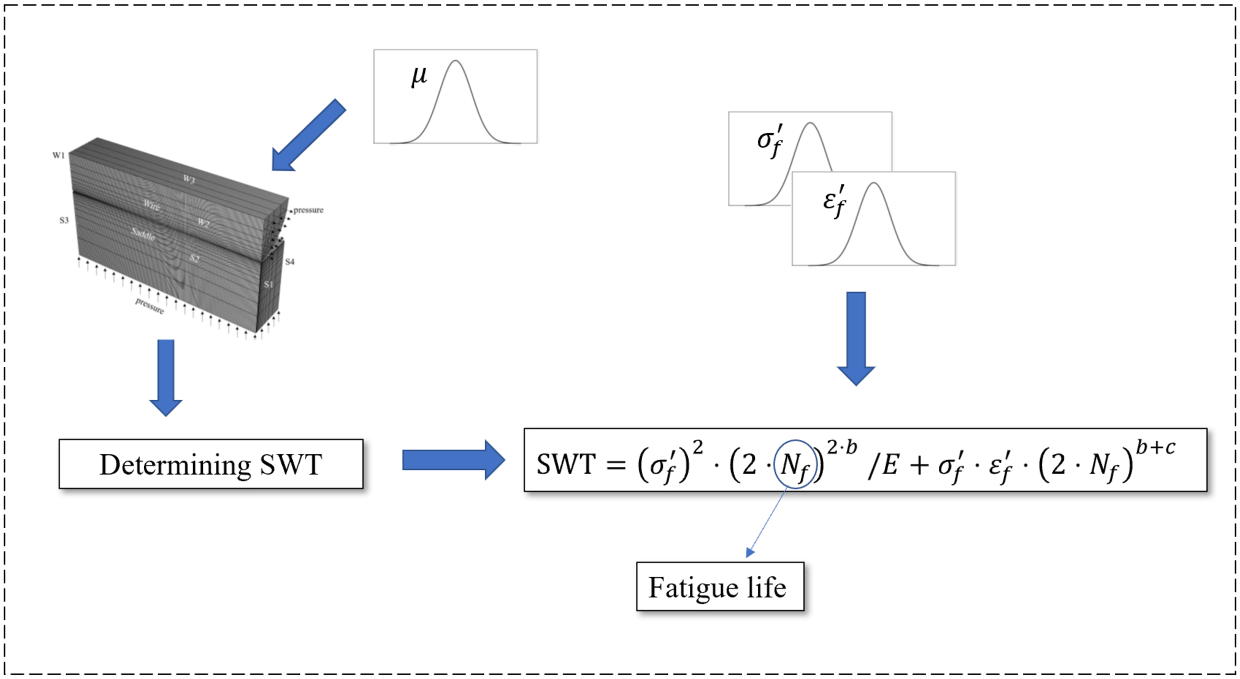


Figure 6-7 The effect of variables in fatigue life estimation of cables.

Similarly to what was done with the fretting maps developed in Chapter 4, it was decided to find a function/map between the coefficient of friction and the SWT parameter. To save analysis time, the quadrature points for COF that were analyzed for the MDRM approach and the results for the upper bound-lower bound COF that were presented in the deterministic part were used to find a function/map between the COF and the SWT parameter. Therefore, an alternative approach based on MCS and a "fretting map" relating the COF and SWT parameter was used. A schematic view of this framework is shown in Figure 6-8.

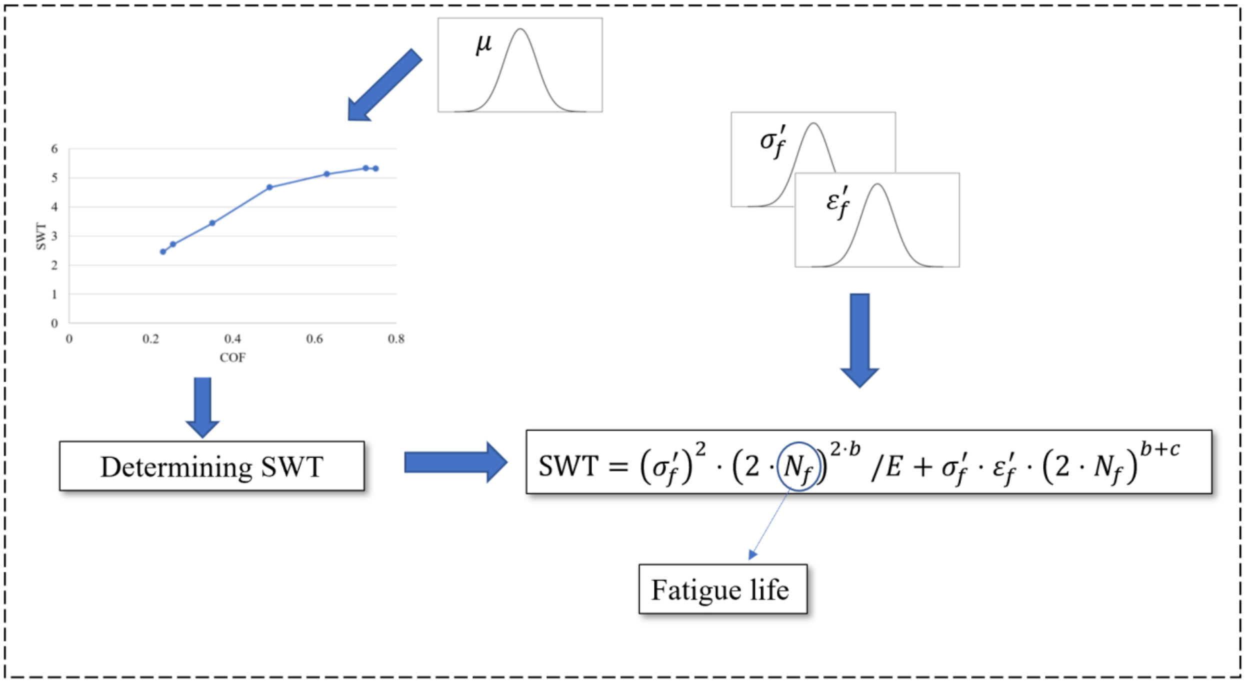


Figure 6-8 MCS along with fretting map framework

6.2.4 Probabilistic analysis results

In the following paragraphs sample of the results are discussed for Test 7 with a contact force of 3 kN and a stress range of 200 MPa; this was the critical point for the tests at 200 MPa. The same analysis was performed for all tests for galvanized and bare wires (see Appendix E).

First, the input grid of the MDRM is shown in Table 6-3. The MDRM results based on different numbers of terms are listed in Table 6-4 and shown in Figure 6-9. It can be seen that the results for $m > 2$ are very close. Another analysis was performed using MCS along with the COF vs. SWT map. It can be seen that the PDF of the fatigue life is transitioning to a bimodal PDF when MCS is used. However, it is not very considerable for this case. Bi-modal PDFs were observed in the analysis for some of the tests (see Appendix E). This behaviour can be explained by looking at the SWT versus COF maps for each test. For some of the tests, the SWT has an almost linear relationship with COF up to a point and then a horizontal line or second line with a smaller slope can be observed. This typically happens when there is a higher uncertainty in the COF. All in all, comparing the CDFs and 95th percent survival probability for all tests based on MDRM and MCS shows very little difference (less than 2% for 95th percent S.P. in most cases, and 5% in the most critical one with high uncertainty in COF). A higher difference was seen when the lower values of CDF were compared in cases with bimodal PDFs.

Table 6-3 Input gird for MDRM analysis of Test #7.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.3592	2183	1.99	3.324	1994574	6.300
2	0.4346	2183	1.99	3.997	505279	5.704
3	0.5450	2183	1.99	4.672	163512	5.214
4	0.6554	2183	1.99	4.893	118433	5.073
5	0.7308	2183	1.99	4.799	135607	5.132
6	0.5450	1890	1.99	4.672	26237	4.419
7	0.5450	2037	1.99	4.672	64498	4.810
8	0.5450	2180	1.99	4.672	160679	5.206
9	0.5450	2333	1.99	4.672	424755	5.628
10	0.5450	2515	1.99	4.672	1292231	6.111
11	0.5450	2183	1.2477	4.672	157459	5.197
12	0.5450	2183	1.5840	4.672	160218	5.205
13	0.5450	2183	1.9650	4.672	163310	5.213
14	0.5450	2183	2.4376	4.672	167103	5.223
15	0.5450	2183	3.0948	4.672	172299	5.236

Table 6-4 MDRM analysis results for Test #7

Moments	Entropy	i	0	1	2	3	4
m=1	1.9805	λ_i	1.894416	6.5868E-12			
		α_i		13.3172			
m=2	0.7242	λ_i	240.1839	-193.3622	27.2879		
		α_i		0.4426	1.0775		
m=3	0.7269	λ_i	1.74E+02	23.9058	24.0061	-133.3021	
		α_i		-2.8276	1.1449	0.5548	
m=4	0.7101	λ_i	497.6401	-243.0761	-407.1558	225.7778	74.8736
		α_i		0.4329	0.2657	0.6180	-1.7160

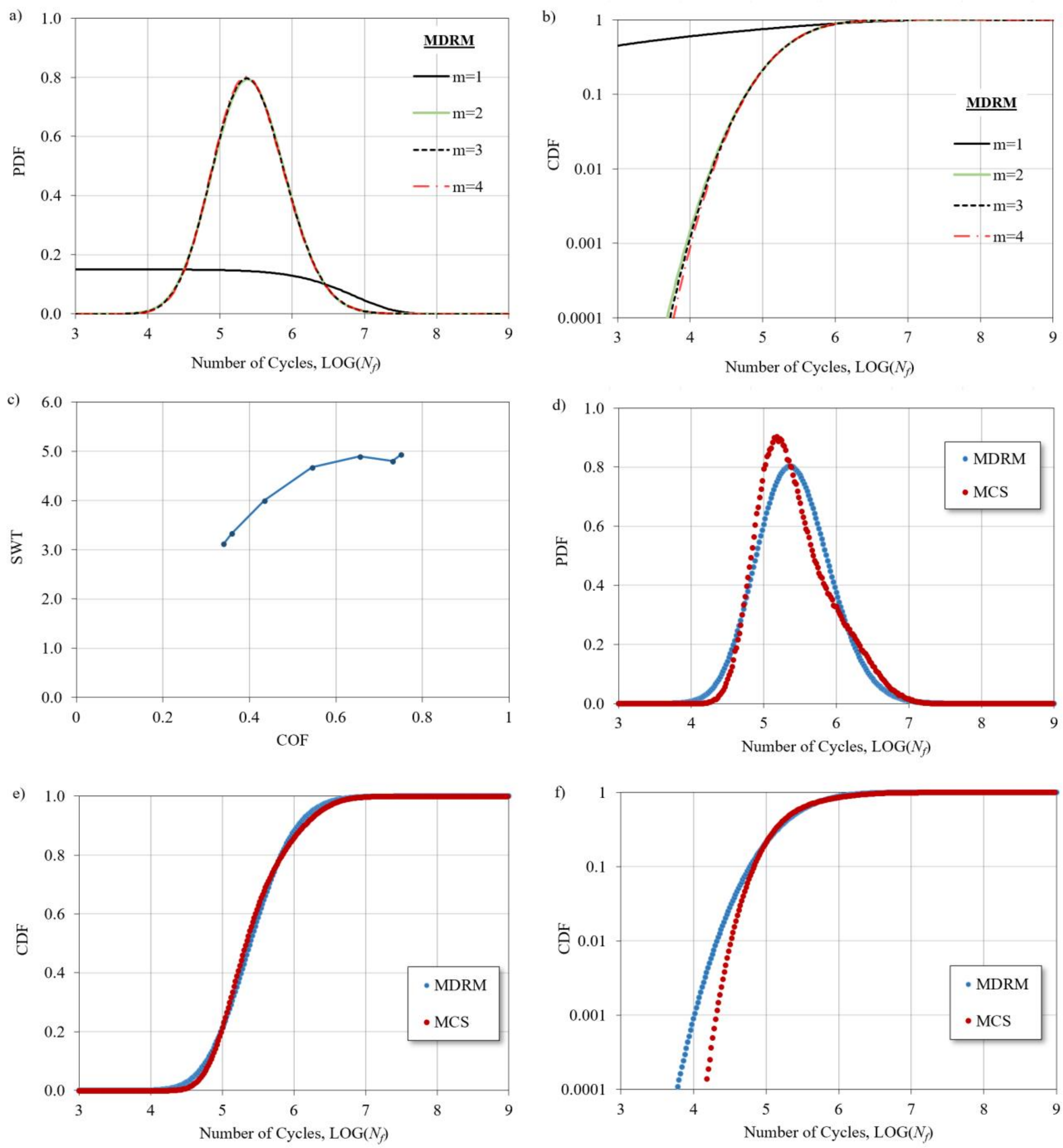


Figure 6-9 Probabilistic analysis for Test #7: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparison of MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on a logarithmic scale (f).

Figure 6-10 shows the results of probabilistic analysis over the test results for galvanized and bare wire. Looking at the results for galvanized wire, it can be seen that the tests were predicted reasonably well and all of them lie above the 95th percent survival probability. Looking at the results for the bare wire, it can be seen that the results are highly conservative in comparison with the test results. As discussed previously, this is due to the challenges in finding the material properties to use in the analysis due to defects inside the bare wire.

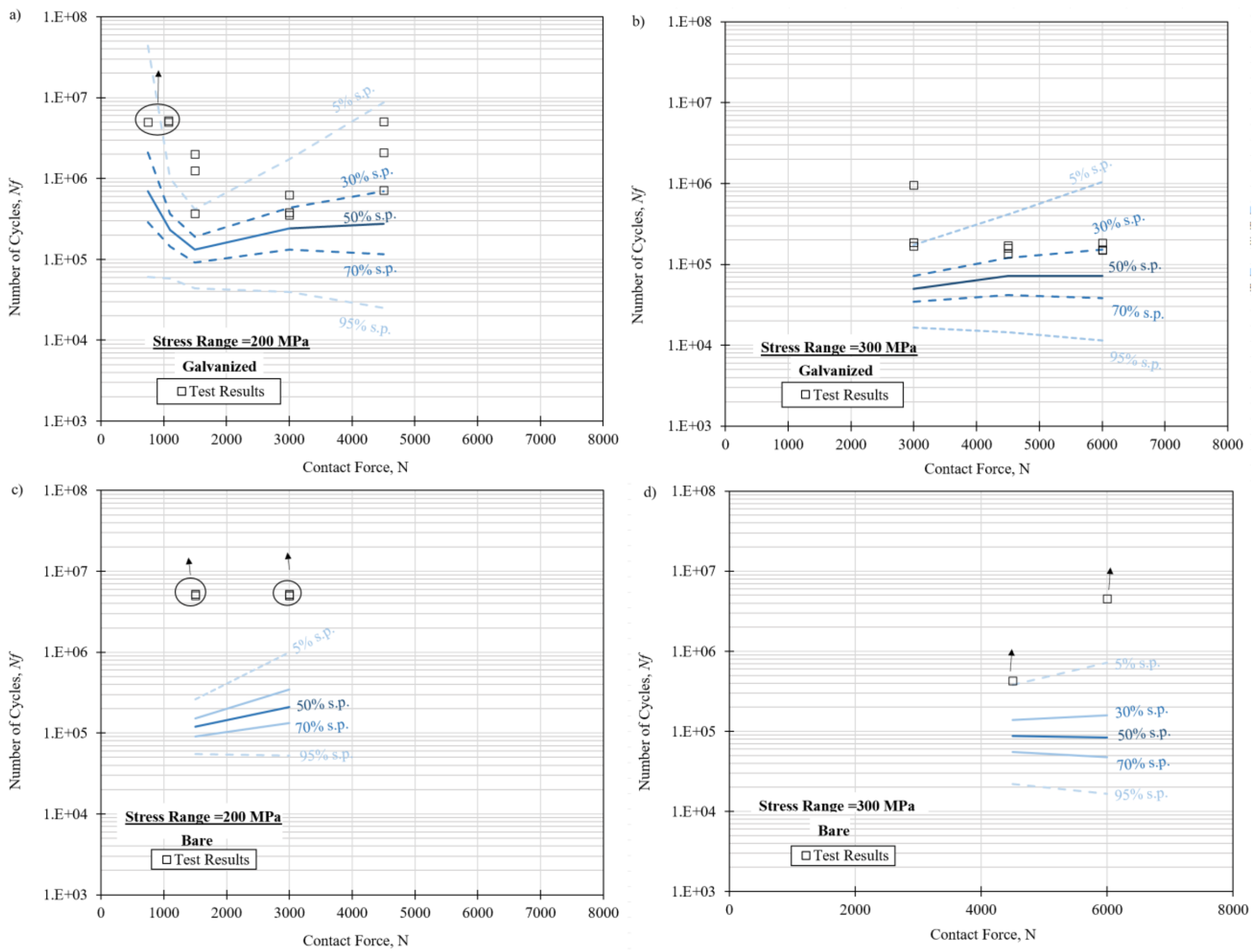


Figure 6-10 Different survival probabilities versus test results.

It is argued that the current analysis was performed based on the knowledge of the upper/lower bound for COF from the tests. However, a designer/researcher might not have access to these numbers. In this case, perhaps a more acceptable approach might be using a conservative assumption for COF and fixing it at the upper bound or considering a uniform distribution for the COF close to the upper bound. Here, it was decided to repeat the analysis using a fixed value for COF at the upper bound and only consider the effect of material variability. Figure 6-11 shows the results based on this analysis. As expected, by making a conservative assumption like the one made here, the survival probabilities decrease. However, the change in the 95% survival probability is limited. It should be noted that this is perhaps a good assumption for real applications since the ratio of frictional force to contact force is typically high in critical points and COF equal to the upper bound can be a good assumption.

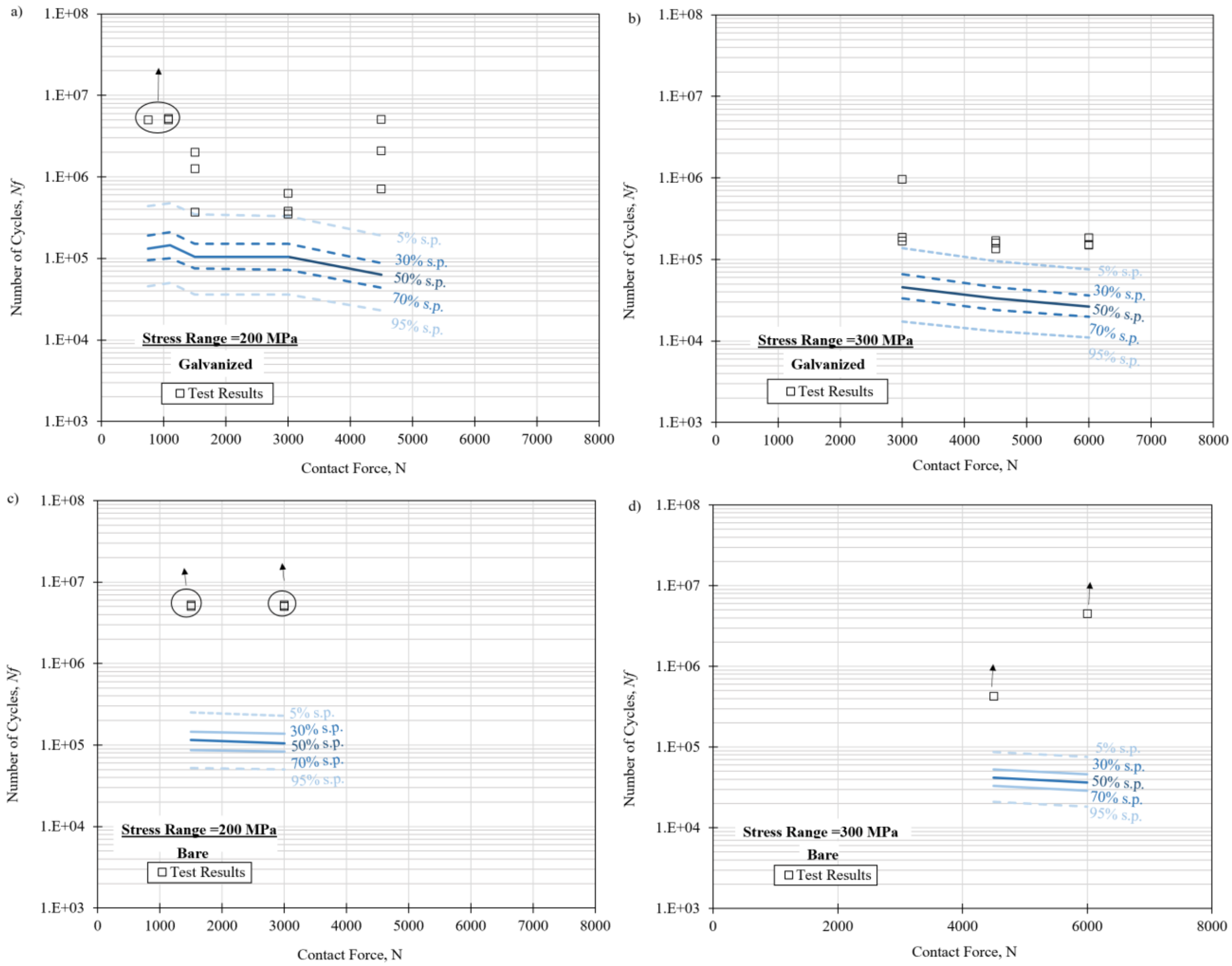


Figure 6-11 Different survival probabilities versus test results assuming a constant COF of 0.75.

7. Analysis of Fretting Fatigue Tests Based on Fracture Mechanics

The approach employed in this thesis until this point to evaluate the fretting fatigue life of cables was based on multiaxial stress analysis along with the SWT parameter. However, the fretting fatigue tests performed in Chapter 5 showed that the fretting fatigue life can significantly depend on the possible defects at the surface of the wires, as evidenced by the lower fatigue life observed for the galvanized wire. The Coffin-Manson parameters do not consider the effects of defects at the surface on fatigue performance. The study in Chapter 5 showed that the bare wire has some internal pores. However, it is very hard at the surface. On the other hand, the galvanized wire has surface defects due to the galvanizing process, which may be impacting fretting fatigue life. With this in mind, while the previously presented multiaxial stress analysis has been shown to lead to good fatigue life predictions, it appears that this approach may be limited by its inability to reveal trends in the test results due to differences in the surface qualities of the different wire types. For this reason, it was decided to evaluate the fretting fatigue life of the wires using an alternative linear elastic fracture mechanics (LEFM)-based approach, wherein surface defect size can be explicitly considered. In this approach, the fatigue life is determined based on the propagation life from an initial small crack. Fretting fatigue analyses based on LEFM and SBFM have been recently discussed in Antunes et al. (2017) and do Rêgo et al. (2018) respectively.

7.1 Methodology

Linear elastic fracture mechanics (LEFM) has been widely used in different engineering fields. As summarized in Chapter 2, several researchers have employed LEFM in fretting fatigue problems. In the current study, the Paris-Erdogan crack growth law is used to evaluate the fretting fatigue life of wires:

$$\frac{da}{dN} = C \cdot (\Delta K)^m \qquad \text{7-1}$$

where a is the crack length, N is the number of cycles, C and m are Paris law constants, and ΔK is the stress intensity factor (SIF) range, The SIF range can be calculated as follows:

$$\Delta K = K_{max} - K_{min} \qquad \text{7-2}$$

where K_{max} and K_{min} are the maximum and minimum stress intensity factors (SIFs). For simple problems, stress intensity factors can be determined using the following equation:

$$K = S \cdot \sqrt{\pi \cdot a} \cdot Y \qquad \text{7-3}$$

where S is the remote stress, a is the crack length, and Y is the product of a series of geometry correction factors. Y factors are usually known for simple geometries and typical loading cases (e.g., tension and bending). However, for more complicated problems other methods are needed. For this reason, the current work employs weight functions to determine stress intensity factors. The stress intensity factors can be determined using weight functions as follows:

$$K = \int_0^a \sigma(x) m(x, a) dx \tag{7-4}$$

where $\sigma(x)$ is the stress distribution along the crack face, which can be determined using FE analysis of the problem, and $m(x, a)$ is the weight function for the points along the crack face. Based on Bueckner (1970) and Rice (1972), given a reference intensity factor, K_r, for a reference stress system, crack opening displacement field of the reference system, u_r, and the generalized elastic modulus, κ, the weight function can be determined as follows:

$$m(x, a) = \frac{\kappa}{K_r} \cdot \frac{\partial u_r}{\partial a} \tag{7-5}$$

While stress intensity factors can be found in the literature for different loading systems. The crack opening displacement fields are typically not reported. Therefore, a generalized form of the weight function is employed, and the unknowns of the weight function are determined using reference stress intensity factors, reference loading, and Equation 7-4. The following two general forms of weight functions have been widely used in the literature (Niu and Glinka 1987, Niu and Glinka 1990, Fett et al. 1987, Sha and Yang 1986, Glinka and Shen 1991):

$$m_{F1}(x, a) = \frac{2}{\sqrt{2\pi(a - x)}} \sum_i \left[1 + M_i \left(1 - \frac{x}{a} \right)^{\frac{i}{2}} \right] \tag{7-6}$$

$$m_{F2}(x, a) = \frac{2}{\sqrt{2\pi(a - x)}} \sum_i \left[1 + M_i \left(1 - \frac{x}{a} \right)^{i} \right] \tag{7-7}$$

It should be noted that only one of the general forms of weight functions should be used in the calculations. However, for comparison purposes, both are first evaluated. In these equations, the only unknows are the $M_i s$. These unknowns can be determined using reference stress intensity factors for specific loading cases from the literature and Equation 7-4. With more reference stress intensity factors, the number of terms in these equations increases.

Cracks start to grow when the stress intensity range is over a threshold. Also, due to crack closure, a portion of the ΔK is effective in crack propagation. Therefore, a modified version of the Paris-Erdogan crack growth law equation was used as follows:

$$\frac{da}{dN} = C\left(\Delta K_{eff}^{m} - \Delta K_{th}^{m}\right) \qquad 7\text{-}8$$

where ΔK_{eff} is the effective stress intensity factor range and ΔK_{th} is the threshold SIF range. The effective SIF range can be calculated as follows (Kurishara et al. 1986):

$$\Delta K_{eff} = U\Delta K \qquad 7\text{-}9$$

$$U = \frac{1}{1.5 - R} \qquad 7\text{-}10$$

where U is the effective stress intensity ratio and R is the stress ratio or the ratio of the minimum stress intensity factor to the maximum stress intensity factor.

7.2 Weight functions

7.2.1 Determining the unknowns of the weight functions

Stress distribution and weight function are required for stress intensity factor calculation as evident in Equation 7-4. Stress distribution can be determined using an FE model. To determine the unknowns of the weight function, reference stress intensity factors from the literature and their corresponding loading systems are required. Geometry factors presented by (Mahmoud 2007) for tension and bending loading have been widely used for the analysis of high-strength steel wires. Here, these factors are used as references for determining the unknowns of weight functions. These geometric factors for tension, Y_t, and bending, Y_b, cases are as follows (Mahmoud 2007):

$$\gamma_1\left(\frac{a}{D}\right) = 0.7282 - 2.1425\left(\frac{a}{D}\right) + 18.082\left(\frac{a}{D}\right)^2 - 49.385\left(\frac{a}{D}\right)^3 + 66.114\left(\frac{a}{D}\right)^4 \quad (Tension) \qquad 7\text{-}11$$

$$\gamma_2\left(\frac{a}{D}\right) = 0.6218 - 0.4014\left(\frac{a}{D}\right) + 0.1127\left(\frac{a}{D}\right)^2 + 4.9954\left(\frac{a}{D}\right)^3 \quad (bending) \qquad 7\text{-}12$$

As two reference SIFs are available, the general weight functions with three terms were used:

$$m_{F1}(x,a) = \frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_2\left(1-\frac{x}{a}\right)\right] \quad 7\text{-}13$$

$$m_{F2}(x,a) = \frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{1} + M_2\left(1-\frac{x}{a}\right)^{2}\right] \quad 7\text{-}14$$

Two equations can be written for each general weight function and the unknown can be found:

$$K_1 = \int_0^a \sigma_1(x)m(x,a)dx \quad 7\text{-}15$$

$$K_2 = \int_0^a \sigma_2(x)m(x,a)dx \quad 7\text{-}16$$

where K_1 and K_2 are the SIFs, $\sigma_1(x)$ and $\sigma_2(x)$ are the corresponding stress distributions. These equations can be written as follows for the first general form of the weight functions, m_{F1}:

$$\sigma\sqrt{\pi a}\gamma_1 = \int_0^a \frac{2\sigma}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_2\left(1-\frac{x}{a}\right)\right]dx \quad 7\text{-}17$$

$$\sigma\sqrt{\pi a}\gamma_2 = \int_0^a \frac{2\sigma\left(1-\frac{x}{r}\right)}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_2\left(1-\frac{x}{a}\right)\right]dx \quad 7\text{-}18$$

where r is the radius of the wire, x is the distance from the surface of the wire, and a is the crack length. The equations can be written for the second general form of the weight functions, m_{F2}:

$$\sigma\sqrt{\pi a}\gamma_1 = \int_0^a \frac{2\sigma}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{1} + M_2\left(1-\frac{x}{a}\right)^{2}\right]dx \quad 7\text{-}19$$

$$\sigma\sqrt{\pi a}\gamma_2 = \int_0^a \frac{2\sigma\left(1-\frac{x}{r}\right)}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{1} + M_2\left(1-\frac{x}{a}\right)^{2}\right]dx \quad 7\text{-}20$$

Unknowns for m_{f1} and m_{f2} can be determined by solving Equations 7-17 and 7-18 and Equations 7-19 and 7-20 respectively. The steps of determining the unknown are shown in Appendix D. The final results for the unknowns of m_{f1} and m_{f2} are as follows:

For m_{f1}:

$$M_1 = \frac{\gamma_1 \pi}{\sqrt{2}} - \frac{2}{3} M_2 - 2 \qquad 7\text{-}21$$

$$M_2 = \frac{15\gamma_2 \pi r}{\sqrt{2}a} + \left(\frac{15}{2} - \frac{15r}{a}\right)\left(\frac{\gamma_1 \pi}{\sqrt{2}}\right) + 5 \qquad 7\text{-}22$$

For m_{f2}:

$$M_1 = \left(\frac{3}{2}\right)\left(\frac{\gamma_1 \pi}{\sqrt{2}} - \frac{2}{5} M_2 - 2\right) \qquad 7\text{-}23$$

$$M_2 = \frac{175\gamma_2 \pi r}{8\sqrt{2}a} + \left(\frac{35}{4} - \frac{175r}{8a}\right)\left(\frac{\gamma_1 \pi}{\sqrt{2}}\right) + \frac{35}{3} \qquad 7\text{-}24$$

7.2.2 Numerical integration using weight functions

Given the stress field, $\sigma(x)$, and the shape function, $m(x,a)$, the stress intensity factor can be numerically determined using the following equation. Different numbers of points can be used for numerical integration. Here the crack length was divided into 1000 points.

$$K = \int_0^a \sigma(x) m(x,a) dx \qquad 7\text{-}25$$

$$= \frac{a}{1000}\left[\frac{1}{2}\sigma(0)m(0,a) + \frac{1}{2}\sigma(0.9999a)m(0.9999a,a) + \sum_1^{999} \sigma\left(\frac{ia}{1000}\right) m\left(\frac{ia}{1000}, a\right)\right]$$

It should be noted that different numbers of points were used to evaluate the accuracy of the integration. It was found that numerical integration with one thousand points is accurate enough. Also, at the last point, instead of a, 0.9999·a was used as $m(x,a) = \infty \; for \; x = a$, and ∞ should not be used for the whole portion of $a/1000$ at the crack tip, as the change in $m(x,a)$ is great at $x = a$. This numerical integration could be done with fewer points in the region where the shape function is more uniform (form x = 0 to 0.95·a) and more points closer to x = a. However, the computational time was not significant at this stage and a uniform version as shown in Equation 7-25 was used for the whole crack length.

7.2.3 Evaluating determined weight functions and numerical integration

The weight functions are determined based on tension and bending loading and their corresponding stress intensity factors. So, with perfect integration, the results of the integral using the shape functions should be exactly the same as the results from the stress intensity factors based on Mahmoud (2007). In other words, it was decided to compare the following two equations and check if the determined unknowns for weight functions and the numerical integration were working properly.

$$K = \int_0^a \sigma(x) m(x,a) dx \quad \text{7-26}$$

$$K = S \cdot \sqrt{\pi \cdot a} \cdot Y \quad \text{7-27}$$

Figure 7-1 and Figure 7-2 compare stress intensity factor results based on Mahmoud (2008) and those calculated using the weight functions. The relative error was lower than 0.2% in all cases for both tension and bending. These figures show that the unknowns were determined correctly. Also, the numerical integration approach is acceptable.

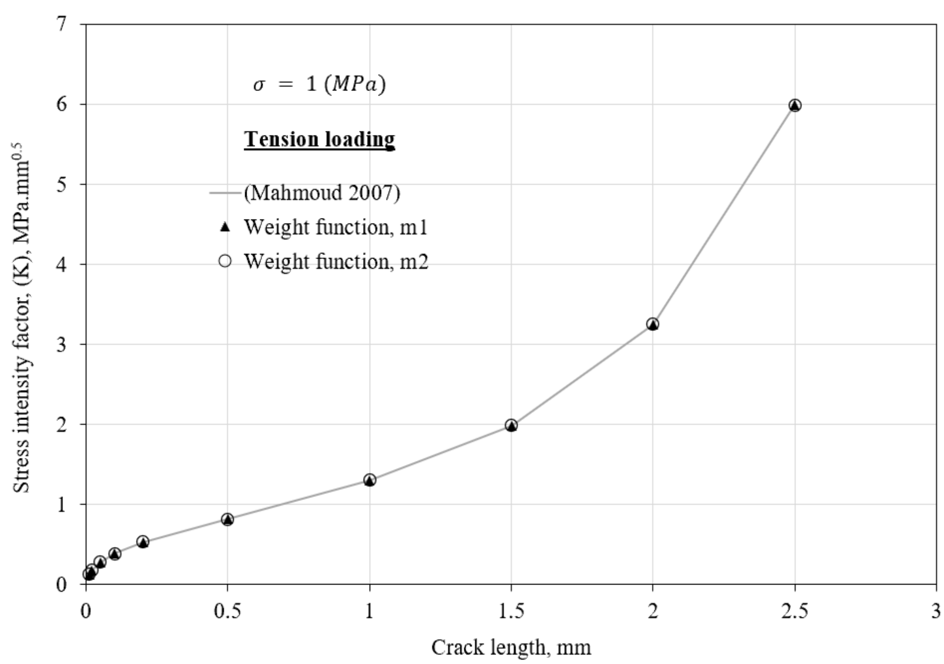


Figure 7-1 SIFs based on weight function results and (Mahmoud 2007) for tension.

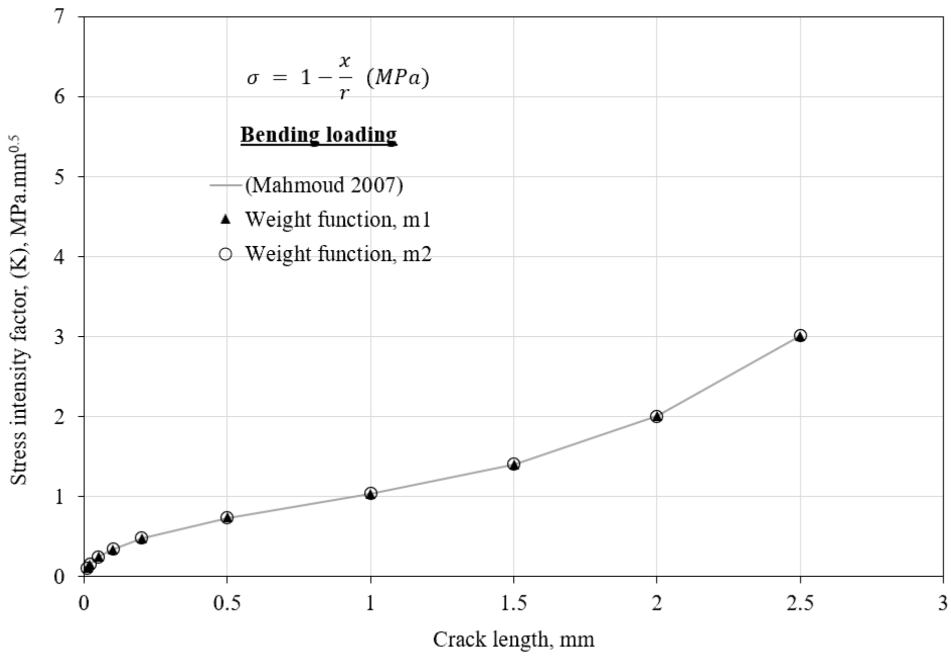


Figure 7-2 SIFs based on weight function results and (Mahmoud 2007) for bending.

7.2.4 Comparison of the weight functions

It was decided to use one of the weight functions for the rest of the current study. The error in the results of the weight functions was minimal for bending and tension. However, to evaluate the possible differences between the weight function results when other types of stress distribution/loading are used, the analysis was done for quadratic and cubic stress distributions as shown in Figure 7-3 and Figure 7-4. The results of both weight functions are close. The difference was lower than 2% and 5% for quadratic and cubic stress distributions. Also, this difference was seen at longer crack sizes which cannot significantly affect the total life. The difference for short cracks (shorter than 1 mm) was lower than 1% in all cases. Stress intensity factors for these loading cases were not found in the literature and the results are not compared with the exact stress intensity factors. However, this analysis showed that the difference between the results based on the studied weight functions is limited. To be consistent in the rest of the study, the first form of the weight function, m_{F1}, was used for stress intensity determination. This form of the weight function has been widely used in the literature (Glinka and Shen 1991).

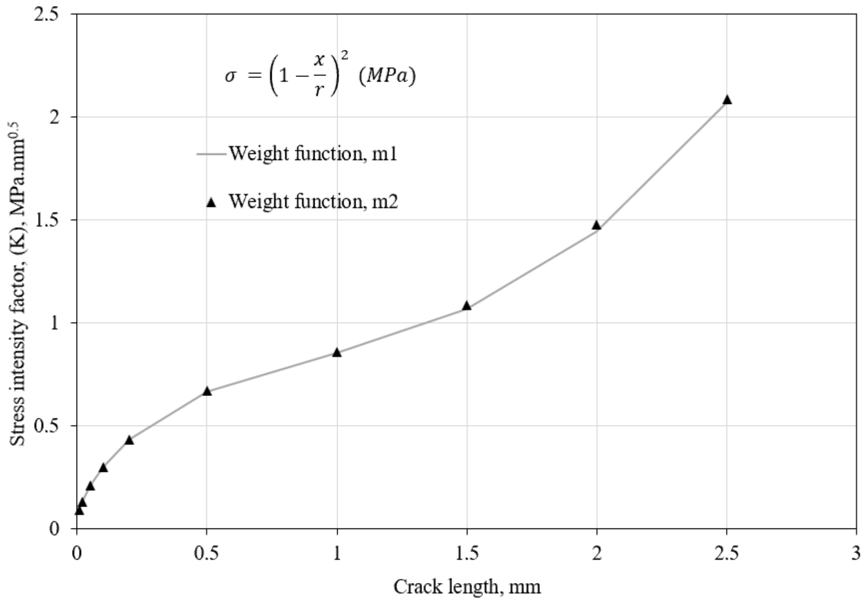


Figure 7-3 Comparing SIFs based on weight functions for quadratic stress distribution.

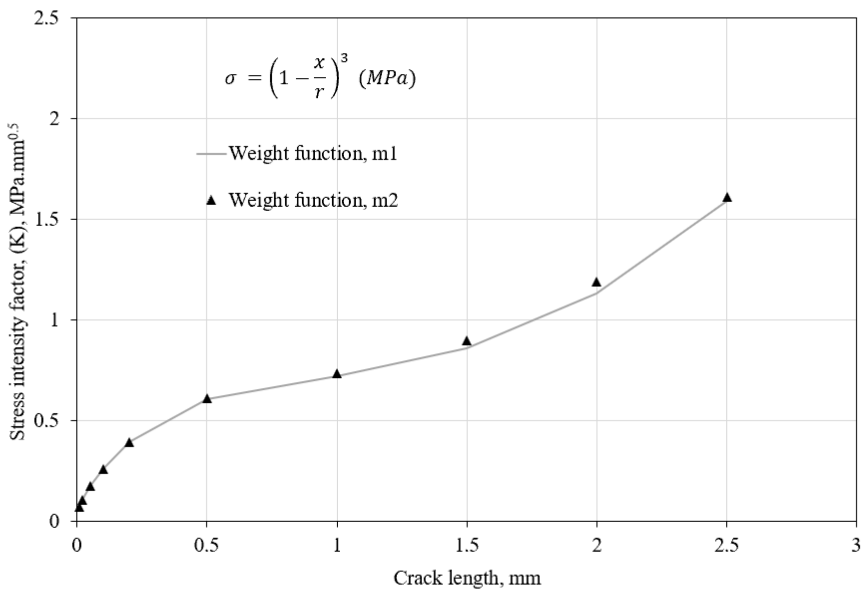


Figure 7-4 Comparing SIFs based on the weight functions for cubic stress distribution.

7.3 Finite element model

To determine the stress intensity factors, stress distribution along the crack length should be determined. To find the stress distribution, an FE model, similar to the one previously presented in Figure 6-1, of the fretting fatigue tests was used. The FE program ABAQUS was employed to model the fretting fatigue tests. In this model, however, elastic material properties with an elastic modulus of 200 GPa and a possion ratio of 0.3 were initially assumed for the saddle/pad (note: the effect of this assumption is studied later). The normal stress distribution was recorded for all points along the contact surface up to the center of the wire. As the exact critical location is not known, the crack propagation analysis was done for all possible vertical cracks.

7.4 Fatigue life prediction

Given the stress distribution from the finite element model and the weight functions determined in the previous sections, the stress intensity factor range was calculated. Given the stress intensity factor range, a numerical version of Paris-Erdogan law was used as follows:

$$\frac{\Delta a}{\Delta N} = C\left(\Delta K_{eff}^{m} - \Delta K_{th}^{m}\right) \tag{7-28}$$

Looking at this equation, a critical crack size should be assumed, above which the wire is assumed to fail. In this work, it is assumed that the length of the crack at failure is equal to the radius of the wire. Another parameter that is required is the initial crack size. This parameter cannot be determined exactly, therefor a parametric study was done for different initial crack sizes from 8 to 14 micrometres. This range is based on the ferrite layers and broken carbide structures at the surface of the galvanized wire reported in Chapter 5. Material properties required for crack propagation analysis were found in the literature. C, m, and ΔK_{th} were considered to be 0.7×10^{-12}, 3.3, 4.24 MPa.m$^{1/2}$ respectively (Zheng et al. 2019, Llorca et al. 1987, Toribio et al. 2009, Lambrighs et al. 2011). As the exact location of the critical point is not known, the analysis was done for all possible vertical cracks along the length of the wire (see Figure 7-5).

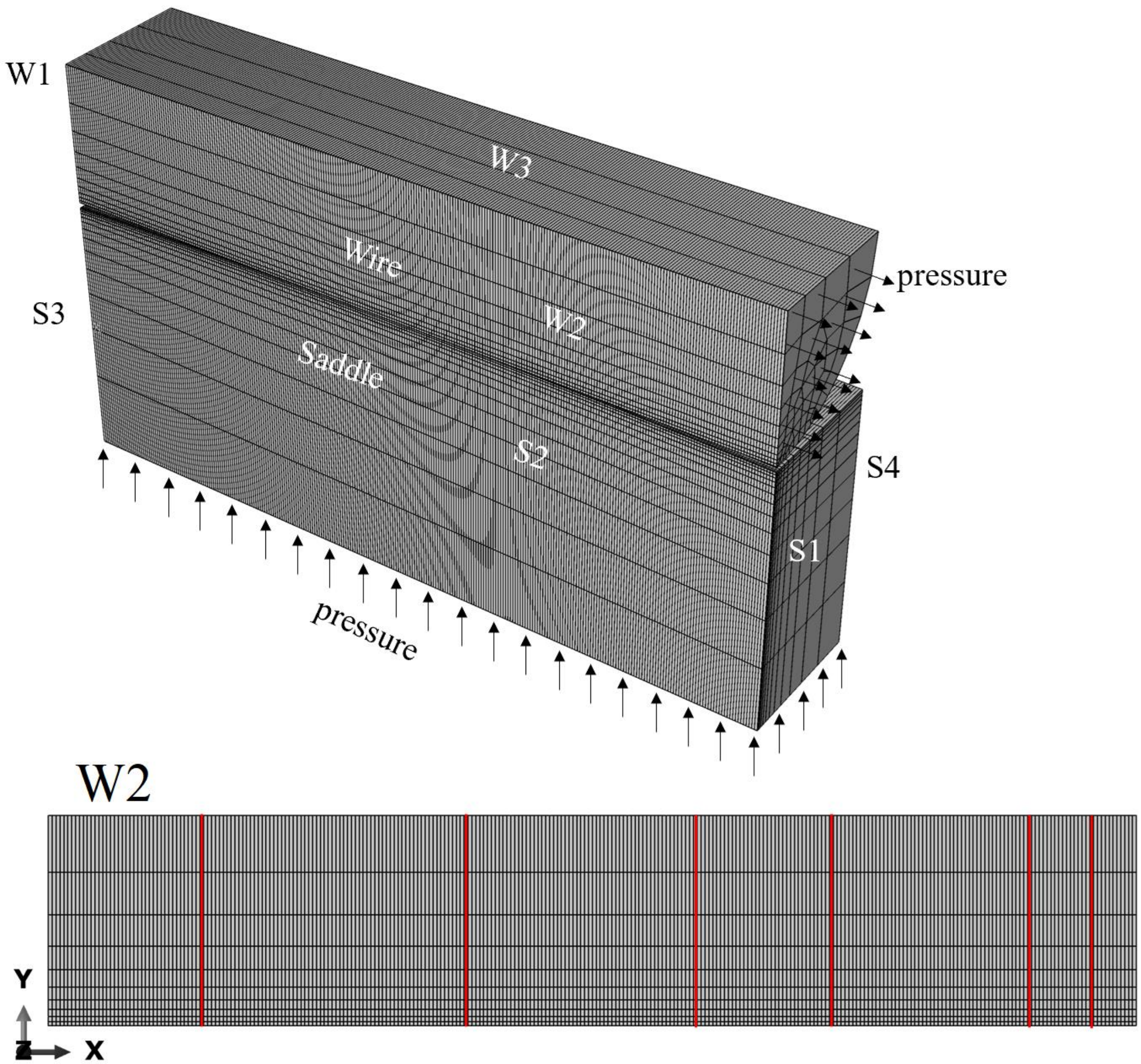


Figure 7-5 Example of evaluated locations for crack propagation.

Sample crack growth paths are shown in Figure 7-5. It should be noted that the analysis was done for vertical cracks every 25 micrometres along the X-axis at the contact surface (Around 560 possible vertical cracks for each test). A python code was used to run the ABQAS for all tests and save the required results. Following that, a MATLAB code was used for fatigue life estimation. Figure 7-6 shows the typical results of the analysis. This figure specifically shows the results for the conditions of Tests # 13-15 in Table 5-1. It should be noted that it was assumed if the number of cycles exceeded 5 million cycles, the code stopped counting and started evaluating the next possible crack. Therefore, at many points, a fatigue life of 5 million is reported. However, the exact fatigue life of these points is higher. This decision was made as the run-out limit in the tests was 5 million cycles and evaluating the points that are not critical is not necessary. As can be seen in Figure 7-6, the points close to the edge of the contact area are critical. This is consistent with the location of failure in the fretting fatigue tests done for this thesis in Chapter 5. All the specimens failed at the points close to the edge of the contact surface. Failure at the edge of the contact is very common in most fretting fatigue problems. It should be noted that at this stage, it was assumed that the stress field is one-dimensional, and only varies along the crack propagation path (i.e, vertical lines shown in Figure 7-5). However, a two-dimensional stress field occurs in reality. Looking at the stress field, it was seen that the stresses are higher along the shown vertical propagation paths in Figure 7-5. Therefore, working with a one-dimensional stress field is a conservative assumption in this case. Additionally, it was seen that the stress variation in the direction normal to the crack growth path was limited, especially at small crack lengths which are associated with most of the fatigue life of the wire.

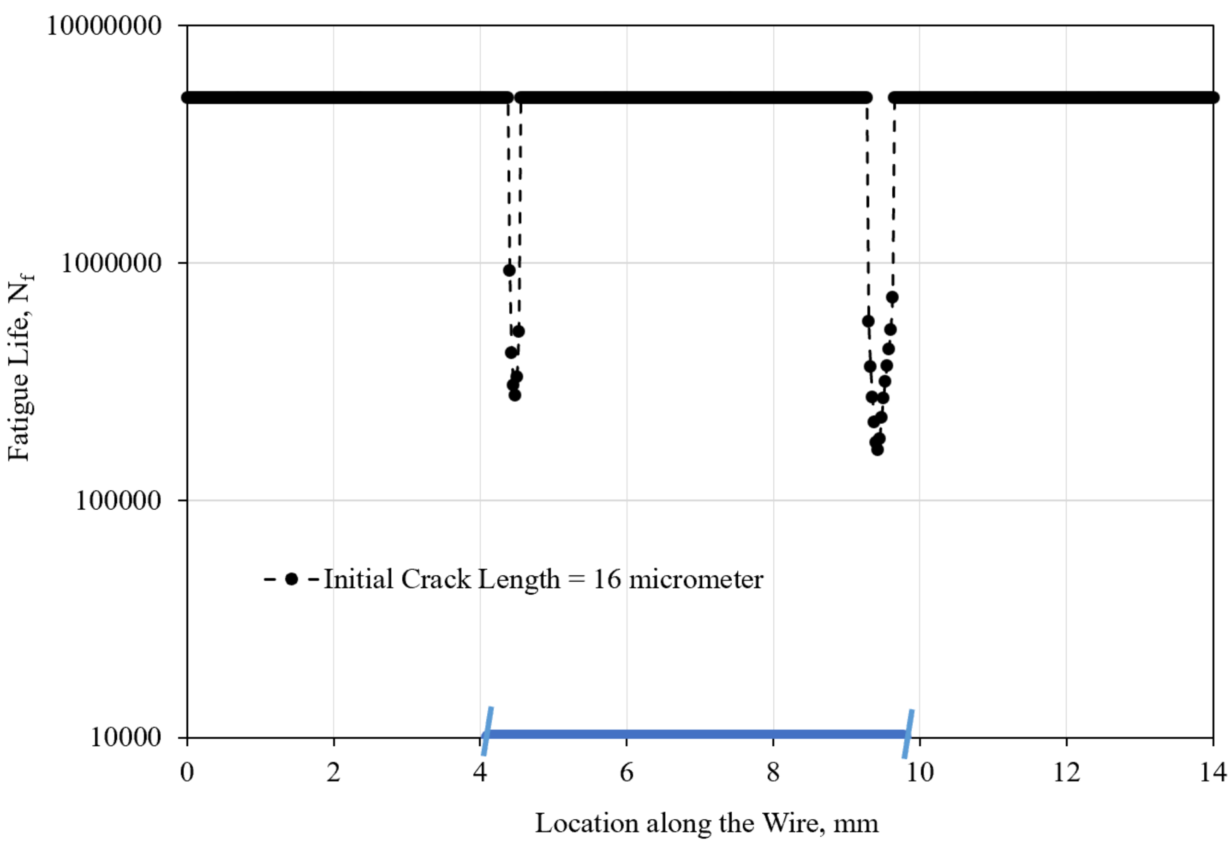


Figure 7-6 Fatigue life based on crack propagation at different locations along the wire for an initial crack length of 16 μm and the conditions of Tests # 13-15 in Table 5-1.

Figure 7-7 compares the results for three different initial crack lengths. Looking at this figure, it can be seen that the location of the critical point is at the edge of the contact area for all cases. As expected, the fatigue life decreases with an increase in the initial crack length. Also, it can be seen that the number of critical points increases with an increase in the crack length, as longer cracks can propagate more easily in comparison with shorter cracks. Failure at the edge of the contact was seen in all fatigue tests, which also was predicted using the SWT parameter.

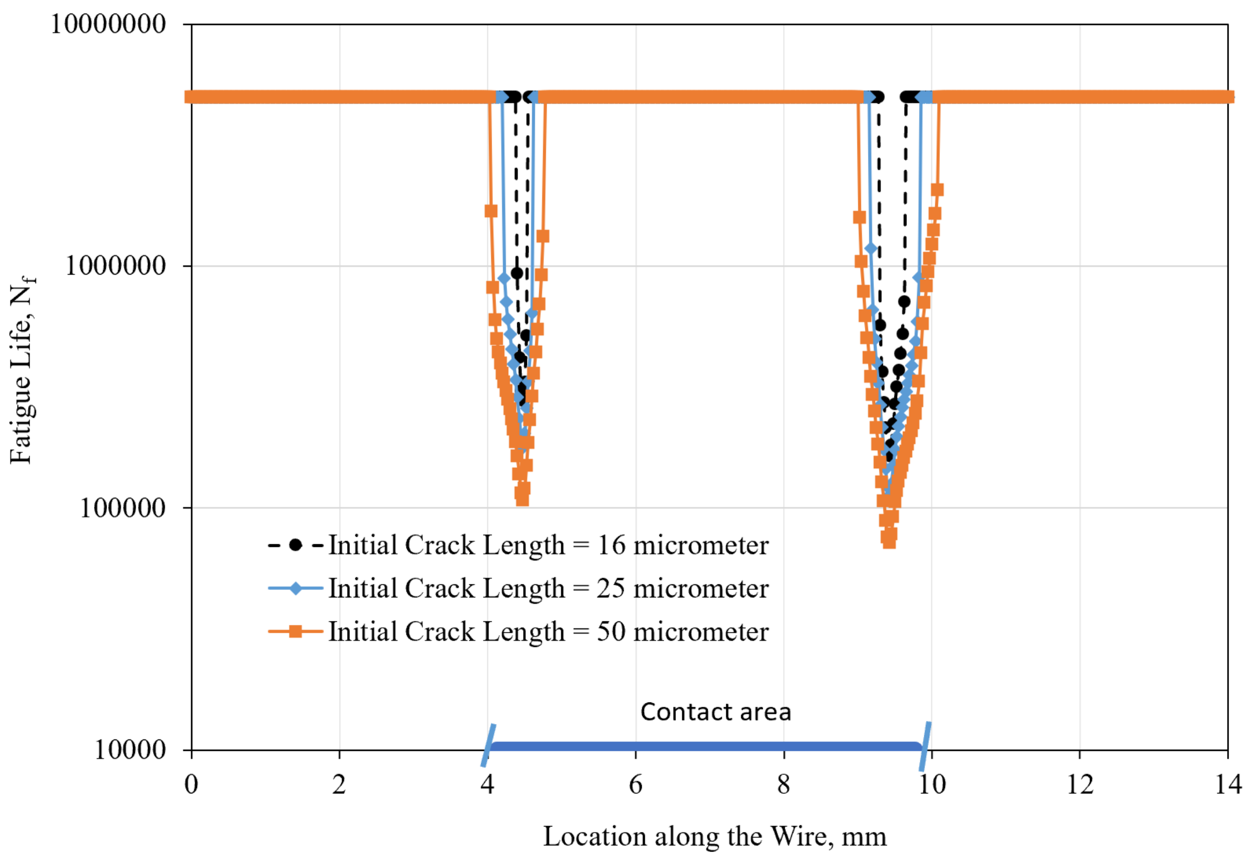


Figure 7-7 The effect of initial crack length on fatigue life along the wire for the conditions of Tests # 13-15 in Table 5-1.

Figure 7-8 shows the fatigue life versus initial crack length for a range of crack lengths from 6 to 100 μm. The analysis was stopped when the fatigue life was over 5 million cycles or when the SIF range was lower than the threshold range. The results rapidly change around an initial crack length of 9.5 μm. After this, the change in the fatigue life is relatively smooth.

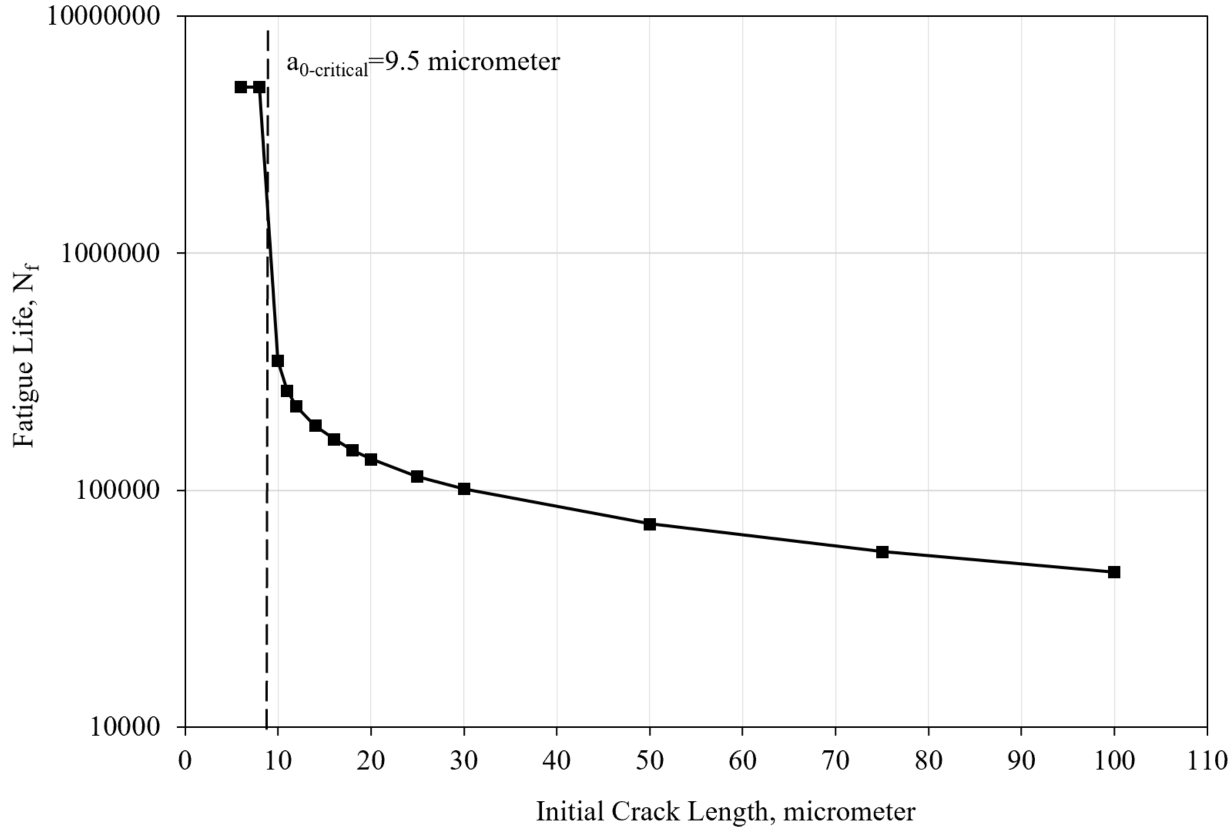


Figure 7-8 The effect of initial crack length on fatigue life for the conditions of Tests # 13-15 in Table 5-1.

7.5 Evaluating the fretting fatigue tests

The analysis was performed for the fretting fatigue tests presented in Chapter 5. The measured slip displacement and contact force and stress range were directly used in the 3D FE model of the contact point. However, as discussed in Chapter 5, the COF cannot be exactly measured and modelled in a stick-slip regime. Therefore, two values representing upper and lower bounds were used for the COF: the gross sliding regime COF, which is equal to 0.75 based on the COF tests presented in Chapter 5, and a quasi-COF, which is the ratio of the frictional force to the contact force. Another parameter that has a high uncertainty is the initial crack length. The affected length at the surface of the galvanized wire by the ferrite layers or broken carbide structures was typically between around 5-15 μm. To cover a range of crack lengths, the analysis was done for four different initial crack lengths of 8, 10, 12, and 14 μm.

Figure 7-9 to Figure 7-12 compares the test results of galvanized wire with the analysis results. First, it can be seen that the results based on quasi-COF are not conservative. They show an upper bound for the fretting fatigue test results. On the other hand, the results based on the gross sliding COF (0.75) are a lower bound of the test results. It can be seen the difference between the predictions based on different COFs gets higher at higher contact forces. This can be explained by the lower difference between the quasi-COF and gross sliding COF at lower contact forces. The exact COF should fall between these two values. Measuring the COF in the stick-slip regime more accurately and using a parameter that considers the effect of both COFs are interesting possible topics to explore in future work. A possible way to consider the effect of both gross sliding COF and quasi-COF might be employing an effective COF as follows:

$$COF_{eff} = (1 - \lambda) \cdot COF_{gross\ sliding} + \lambda \cdot COF_{quasi} \tag{7-29}$$

where λ varies between 0 and 1. Looking at the results in Figures 7-9 to 7-12, it can be seen that the test results are closer to predictions based on the gross sliding COF. Therefore, probably the λ factor should be somewhere between 0.1 to 0.3. More analysis and tests focused on the COF are required to further investigate this possible expression for COF and λ factor.

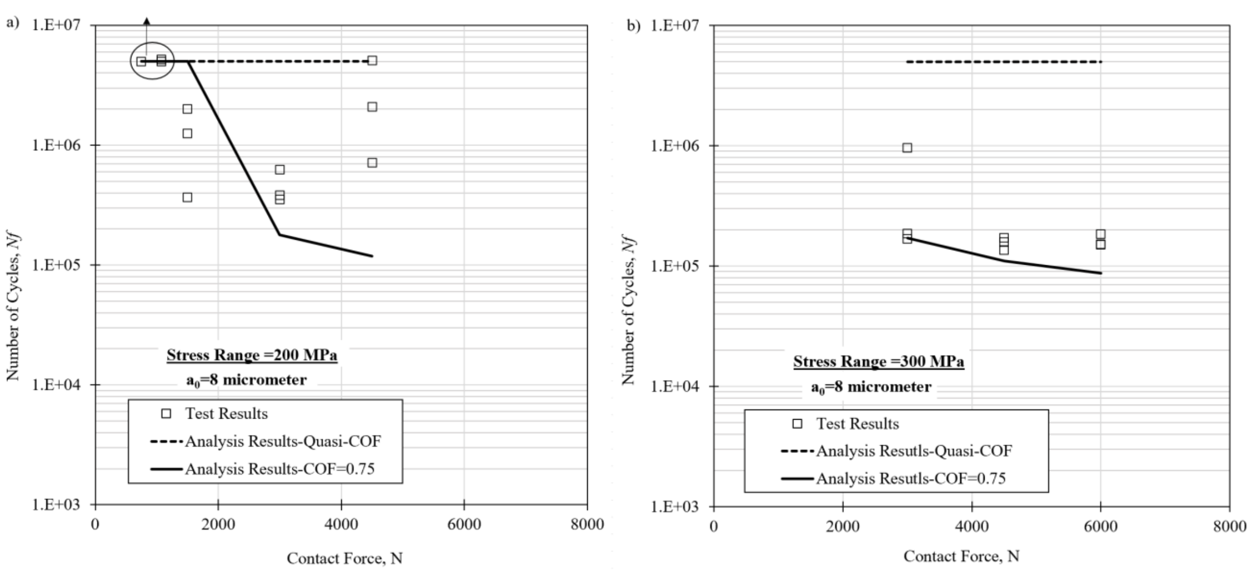


Figure 7-9 Fatigue life prediction of galvanized wires with an initial crack length of 8 µm.

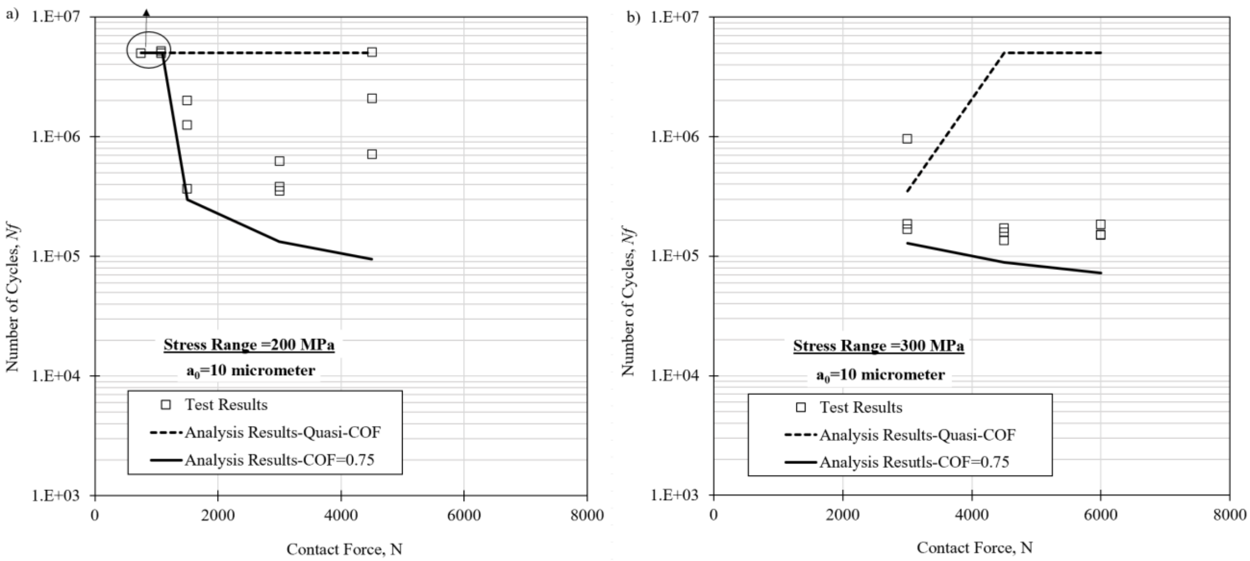


Figure 7-10 Fatigue life prediction of galvanized wires with an initial crack length of 10 μm.

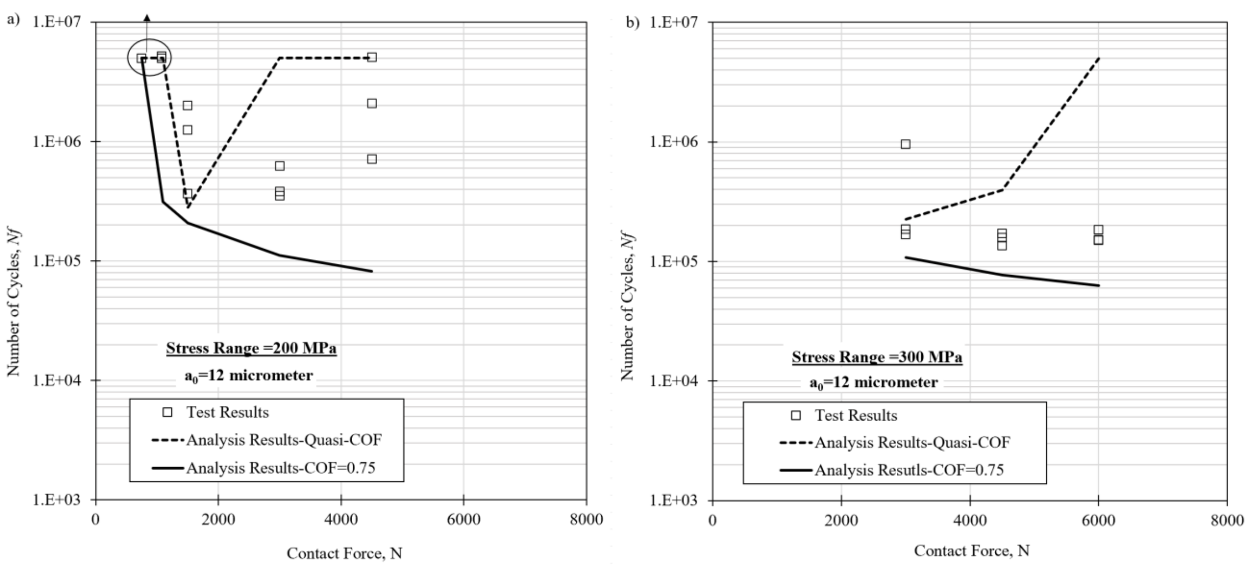


Figure 7-11 Fatigue life prediction of galvanized wires with an initial crack length of 12 μm.

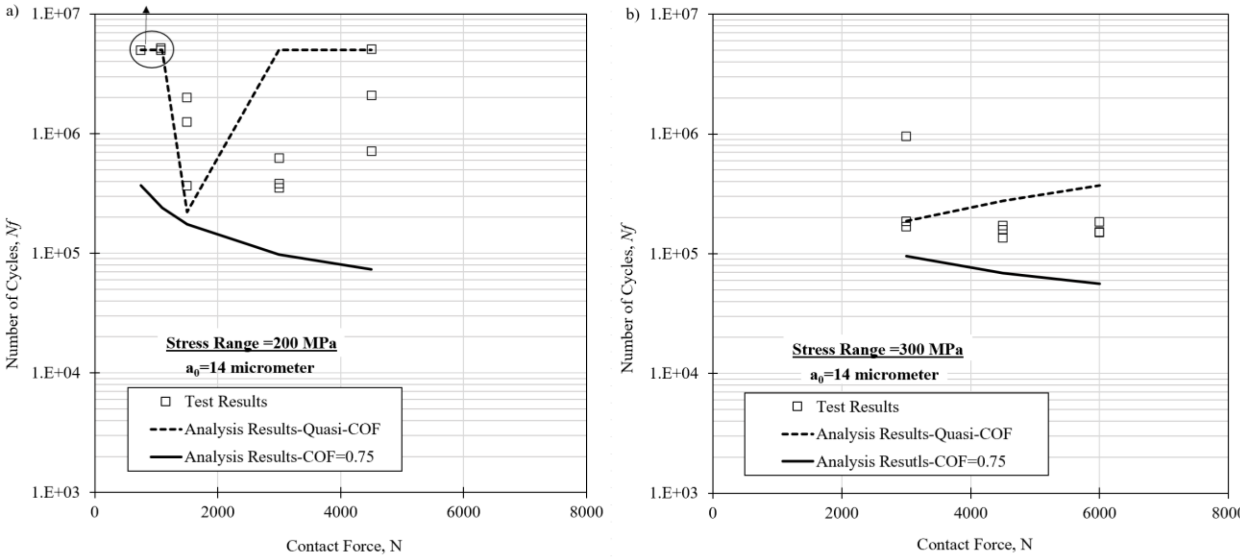


Figure 7-12 Fatigue life prediction of galvanized wires with an initial crack length of 14 µm.

The analysis was primarily performed using elastic material properties for the contacting pads. However, under high contact forces, the elements at the surface of the pads can yield. Considering elastic material properties for the pad is typically a conservative assumption as stresses are higher in an elastic analysis. However, to evaluate the possible effect of considering elastic-plastic properties for the pad on the results, the analysis was repeated by considering an elastic-perfectly plastic material model with an elastic modulus of 200 GPa and yield stress of 500 Mpa for the contacting pads. The results of this analysis are compared with the analysis based on the elastic material model for initial crack lengths of 12 and 14 micrometres in Figure 7-13 and Figure 7-14 respectively. As expected, the fretting fatigue life predictions based on the elastic material model are conservative in comparison with the elastic-plastic material model.

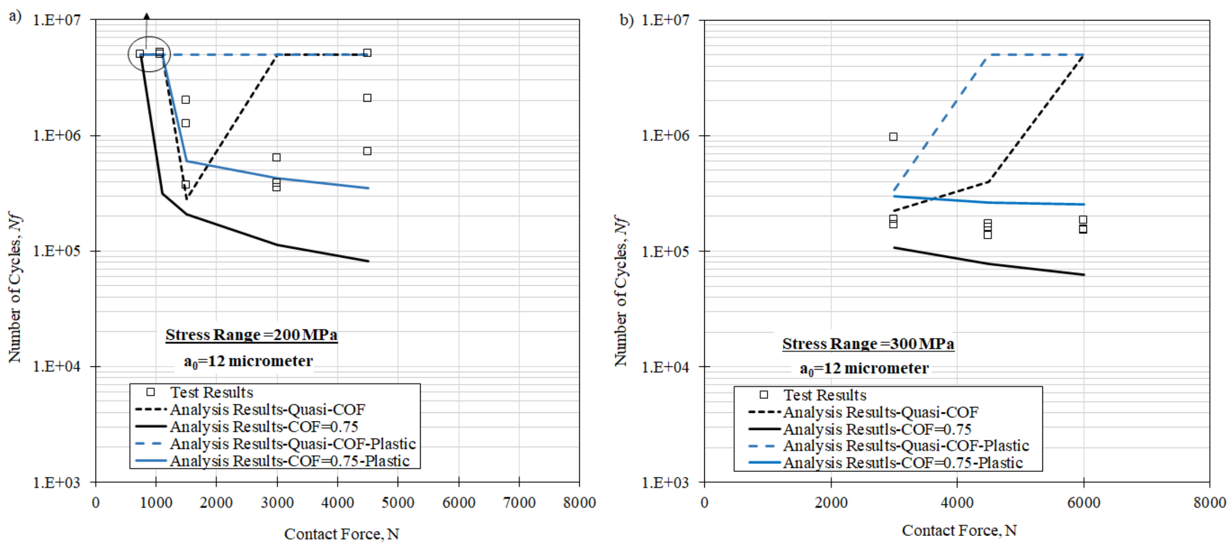


Figure 7-13 Fatigue life prediction of galvanized wires with an initial crack length of 12 μm considering an elastic-plastic material model for pads.

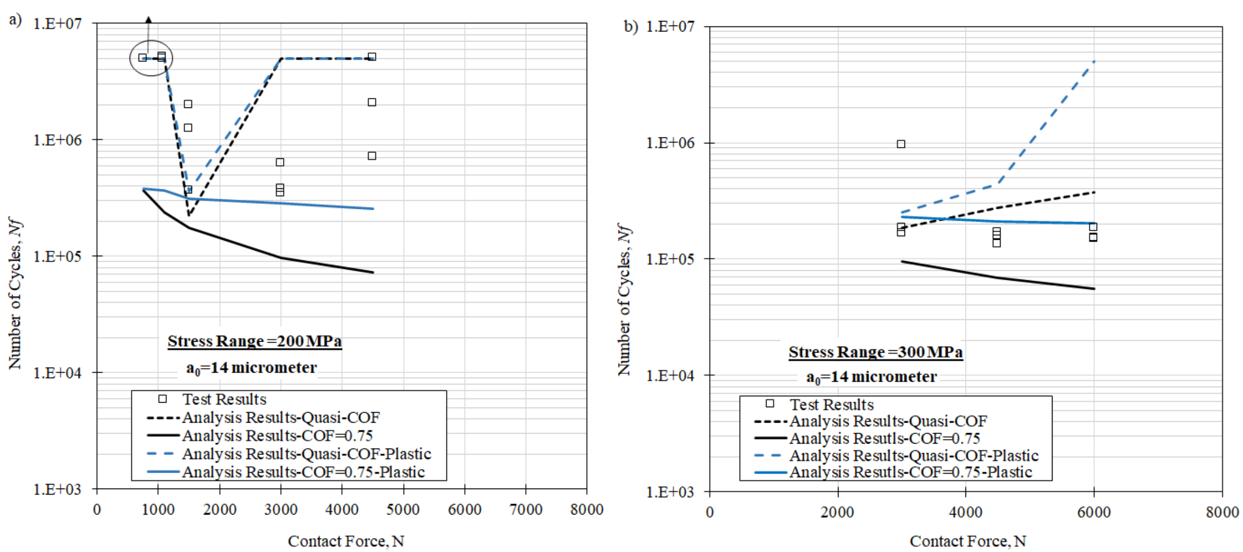


Figure 7-14 Fatigue life prediction of galvanized wires with an initial crack length of 14 μm considering an elastic-plastic material model for pads.

It should be noted that this chapter presents results for galvanized wire only. As discussed in the previous chapter, no failure was observed in fretting fatigue tests of bare wires. Also, no obvious defect was detected at the surface of the bare wires to be considered as an initial crack. The fretting fatigue tests were done at stress ranges of 200 and 300 MPa. The bridge cables should be designed for a stress range of 200 MPa. It seems that further increasing the stress range is required to get fretting fatigue failure with bare wires. However, increasing the stress range to 400 or 500 MPa can cause failure at the grips. Also, tests at those stress ranges are not relevant for bridge structures. They, however, can be performed to better understand the fretting fatigue performance of the bare wires. Even at those stress ranges, the crack propagation framework might not be suitable as very small cracks cannot grow. The best approach to evaluate the fretting fatigue life of bare wires without obvious defects at the surface may be using a mixed crack initiation-propagation approach similar to the one in Navarro et al. (2008) or a small crack fracture mechanics approach, such as strain-based fracture mechanics (SBFM) (Walbridge 2008, Ghahremani and Walbridge 2011, Yekta et al. 2013). It should be noted that the tests and analysis in the current study were for fretting fatigue tests in air. Considering the effect of corrosion is another topic to explore. Corrosion effects might considerably decrease the fretting fatigue life of bare wires in comparison with galvanized wires by causing surface roughness and material loss over time. All in all, it seems that while crack initiation methods are relatively simpler to implement, they cannot properly account for the effect of surface defects. The LEFM framework presented in this chapter provides the possibility of considering the initial defect size as a parameter in the analysis and can be helpful when significant defects can be observed at the surface of components.

8. Conclusions and Future Work

This chapter presents (in Section 8.1) the main results and contributions of the research presented in the current thesis. It concludes with a discussion (in Section 8.2) of possible areas of future work on the topic of fretting fatigue testing and analysis for stay cable saddle systems.

8.1 Conclusions

The main conclusions and contributions of this thesis can be summarized as follows:

- Comparing the analytical solutions in the literature with the results of the 2D FE analysis of a cable over a saddle for the critical parameters at the contact points (contact forces and slip displacements) shows some differences, especially at the points where the cable first meets the saddle. However, the results are very close in the uniform region. Therefore, for the non-uniform region (i.e. the first contact points) either a more complex/advanced approach, such as the 2D FE analysis presented in the current study, should be used or a safety factor should be applied to the analytical solution results.
- Through modifications to the 2D FE analysis, it was shown the effect of wear is higher at the first contact points. Looking at the fretting fatigue life predictions in Chapter 3, it can be seen that the critical point is the first contact point when the wear is not modelled. However, the second contact point becomes more critical when the wear is modelled. This lower fatigue life at the second contact point can be explained by the higher contact force at this point in comparison with the first contact point when the effect of wear is considered, as the wear results in a redistribution of the forces at these contact points. The model with wear effects considered appears to capture more realistically the observed failure mode in large-scale saddle system fatigue tests conducted previously by others.
- Crude MCS was found not to be a practical way to conduct a probabilistic analysis of this problem, as each trial requires a time-consuming nonlinear 3D FE analysis. Therefore, it was shown how more efficient approaches including MCS along with fretting maps and M-DRM, can be used for the current problem both of which require a much smaller number of FE analyses and consequently much lower computational times. The determined CDFs of the fretting fatigue life based on the two presented frameworks were very close to each other.

- It was shown that the concept of fretting maps can be used to develop a practical design tool for stay cable saddle systems, although, because of the high computational time of the FE analysis, generating these maps is time-consuming. Further investigation is therefore recommended to determine the best approach for establishing fretting maps covering a broad enough range of parameters to serve as a generalized design tool.
- A probabilistic sensitivity analysis was done using M-DRM where it was found that the most important parameters affecting the prediction of fretting fatigue life are the bias factors for the contact force and the fatigue strength coefficient of wires. This was the same for both galvanized and bare wires.
- A small-scale fretting fatigue setup was designed, fabricated, and used for fretting fatigue tests on single cable wires. Small-scale test setups are generally more economical and efficient compared to large-scale ones. Details of the design, drawings, challenges, measurements, and effective parameters have been presented.
- The fretting fatigue performance of two cable types, namely bare and galvanized, was evaluated using a small-scale test setup. It was shown how stress range and contact force affect the fretting fatigue life of the wires. Also, other parameters including slip displacement, frictional force, and COF were measured and reported. Similar to plain fatigue tests, fretting fatigue life considerably decreased with an increase in the stress range. The fretting fatigue tests showed that increasing the contact force initially decreases the fretting fatigue life. However, fretting fatigue life can increase at higher contact forces. This was most obvious in the fretting fatigue test results at the 200 MPa stress range. However, a smaller difference was still observable between the test results at the 300 MPa stress range.
- Microhardness, tensile and plain fatigue tests were performed and the properties of both the bare and galvanized wires including microhardness, elastic modulus, yield stress, tensile strength, fracture strain, and Coffin-Manson parameters are reported. It was found that the yield stress of both wires was relatively close. The core material of the galvanized wire had a higher fatigue limit than that of the bare wire. However, the microhardness tests showed that the bare wire was harder in comparison with the galvanized wire.

- Microstructure analysis was done to detect defects in the bare and galvanized wires and allow their possible effect on the fretting fatigue and plain fatigue performance of the wires to be considered. It was found that the bare wires have internal pores. These pores reduced the fatigue limit of the bare wire in comparison with the galvanized wire. On the other hand, the galvanized wires had defects at the surface, while the bare wire had a uniform pearlite structure at the surface. It is believed the defects at the surface of the galvanized wire reduced their fretting fatigue life in comparison with the bare wire.

- Given the defects at the surface of the galvanized wires, another approach based on linear elastic fracture mechanics was used to predict the fretting fatigue life of the wires. Different initial crack lengths from 8 to 14 μm were used to evaluate the effect of initial defect size on the test results. The predictions were relatively close to the test results, especially when the gross sliding COF was used in the analysis.

- The unknowns for two different general weight functions were determined for wires. It was shown that stress intensity factors obtained using these weight functions are very close to each other for quadratic and cubic stress systems. These weight functions can be employed in different linear elastic fracture mechanics (LEFM) analyses of wires subjected to a one-dimensional stress field.

8.2 Future work

The following is a list of possible areas of future work stemming from the presented research on the fretting fatigue testing and analysis for stay cable saddle systems:

- *Large-scale tests of saddle systems to evaluate the critical parameters at the contact points*: More large-scale fatigue tests are needed to evaluate the results obtained using the closed-form equations and the 2D FE analysis. These tests should be done to evaluate the accuracy of these methods and establish bias factors to consider the associated model uncertainties.

- *Fretting fatigue tests focused on the coefficient of friction*: Fretting fatigue tests should be done to evaluate the COF in the stick-slip regime more accurately. The COF can be determined when the wire completely moves along the pads. Fretting fatigue tests cannot be stopped to measure COF under such conditions, as they can cause a high degree of wear and thus alter the fatigue test result. Therefore, several tests focused on COF measurements can be another area of future work. DIC and stepwise increase of axial load are possible approaches to explore for this possible future work.

- *Saddle material effect tests:* In the current study, steel pads were used for the fretting fatigue tests. Other materials (e.g., aluminum and high-performance concrete) can be used for the pads to evaluate the effect of saddle material on the fatigue life of these systems.
- *Implementing other fatigue life prediction approaches:* In this work, a multiaxial stress-based approach based on the SWT parameter, and a linear elastic fracture mechanic approach have been used to evaluate the fretting fatigue life of the cables. Other approaches (e.g., X-FEM, strain-based fracture mechanics) can alternatively be used to study this problem and may lead to better fatigue life predictions, given the capabilities they would enable.
- *Modelling the wear in the 3D FE model:* The Archard equation can be used to model wear in the 3D FE model. The stress and strains of the FE model can be recorded. Then, based on the Archard Equation, elements can be removed after a few thousand cycles and a new model can be created. With this approach, the beneficial effect of wear can be modelled.
- *Crack propagation calculation considering the effect of wear:* There is a competition between crack propagation and wear in fretting fatigue problems. An interesting topic is considering the wear depth in the crack propagation framework. As discussed previously, the fretting fatigue life typically increases with a high degree of wear as wear can remove the small cracks that are propagating at the surface of the wires, thus increasing fatigue life.
- *Variable amplitude loading tests:* Fretting behaviour under variable amplitude loading can be an important issue in the case of saddle supports in bridges. How a saddle performs under loading histories containing a combination of large cycles that cause wear and small cycles that cause fretting fatigue damage is another interesting topic to explore. Recent studies showed that the gross vehicle weight histogram of the trucks in real traffic can significantly affect the fatigue life of elements with short to long influence lines (Chehrazi et al. 2022a, Chehrazi et al. 2022b). Therefore, evaluating the performance of these systems subjected to different real traffic databases can be an interesting future project. Apart from that, simultaneous truck crossings, when trucks follow each other closely or cross the bridge side by side, can significantly increase the stress ranges and decrease the fatigue life of components

(Walbridge et al. 2011, Chehrazi et al. 2022c). Evaluating the possible effect of simultaneous truck crossings on the fretting fatigue performance of cables is another interesting project.

- *Analysis of cable-stayed bridges:* Exploring how a designer might establish the dead/live loads for the design of a cable-stayed bridge would make it possible to ensure the developed fretting fatigue analysis methods and design frameworks can be practically implemented.
- *Advanced M-DRM approach:* An advanced version of M-DRM called MDR-PCE has been developed recently (Zhang et al. 2021). This method is capable of capturing the bi-modal behaviour of a function/structural response. Therefore, employing this method can increase the accuracy of determined PDF/CDFs and can be an interesting future project.

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Appendices

Appendix A

The results of contact forces and slip displacements at the contact points for saddle radii of 1000, 1500 mm are shown in this appendix.

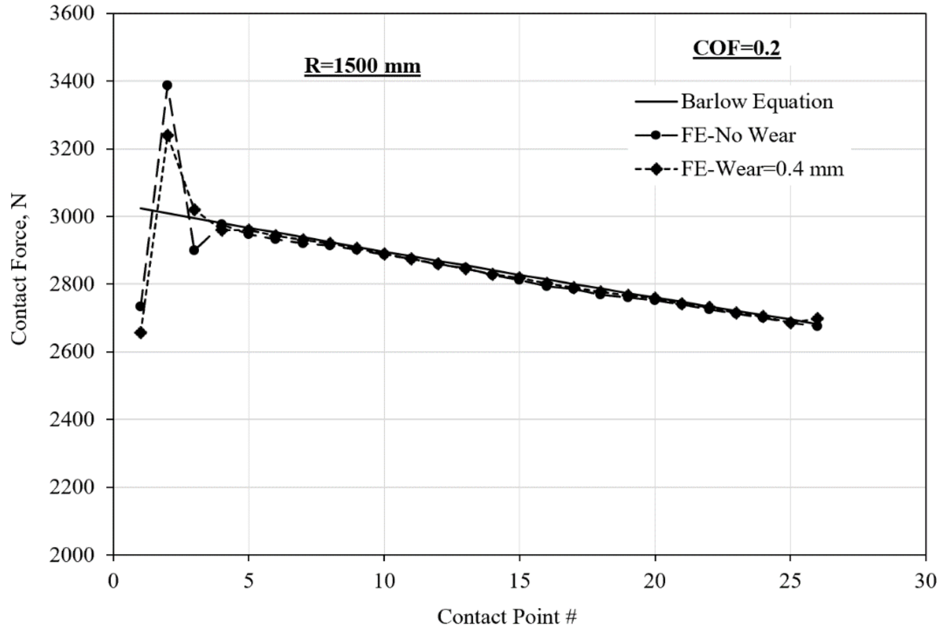


Figure A-1 Contact force results for R = 1500 mm, and COF = 0.2.

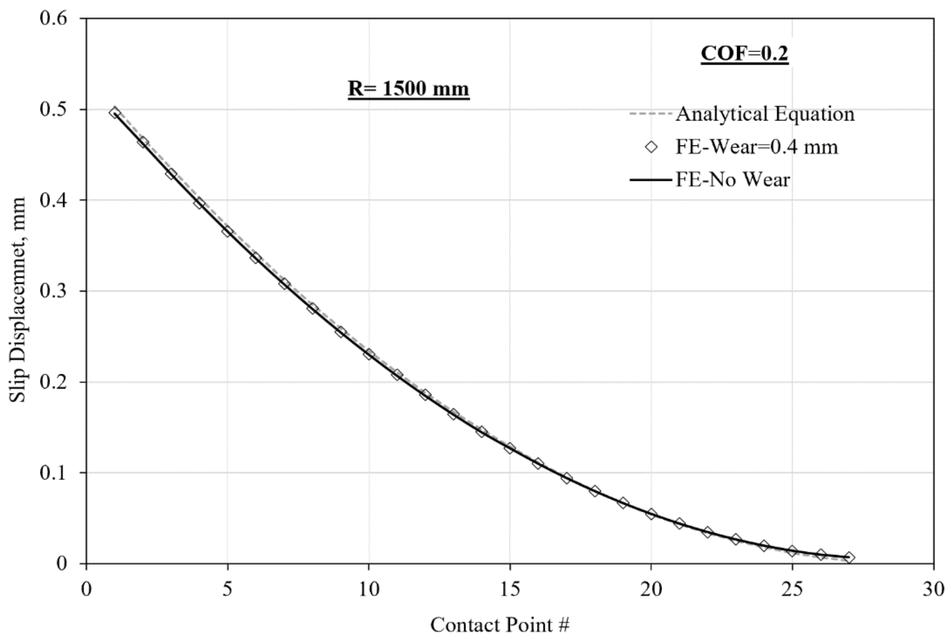


Figure A-2 Slip displacement results for R = 1500 mm, and COF = 0.2.

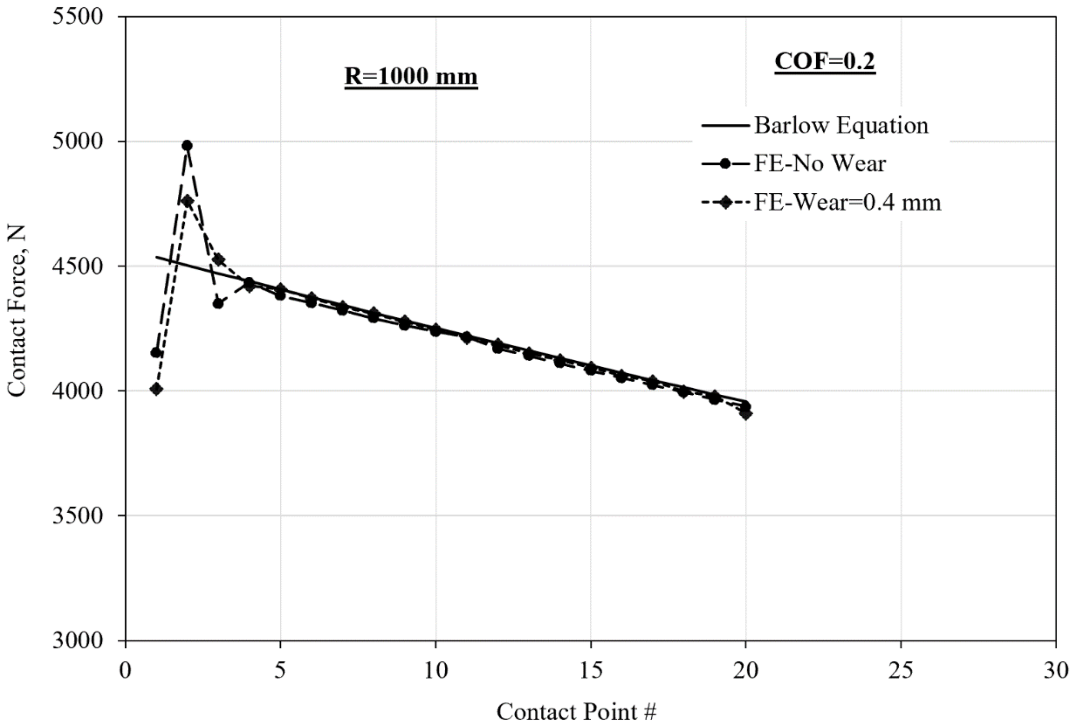


Figure A-3 Contact force results for R = 1000 mm, and COF = 0.2.

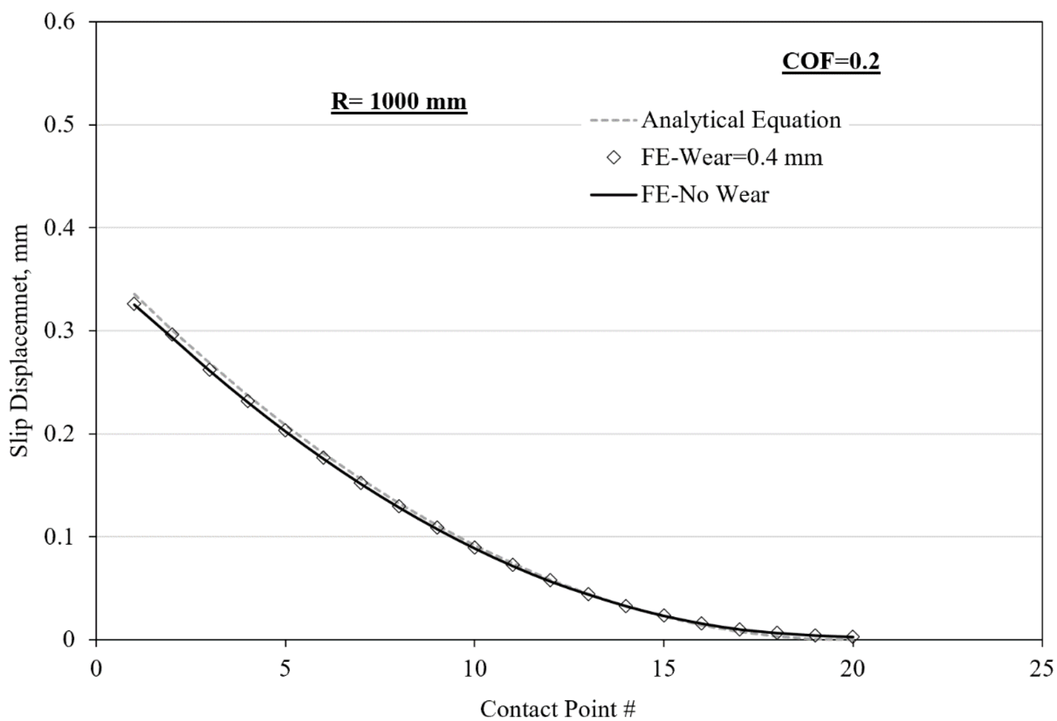


Figure A-4 Slip displacement results for R = 1000 mm, and COF = 0.2.

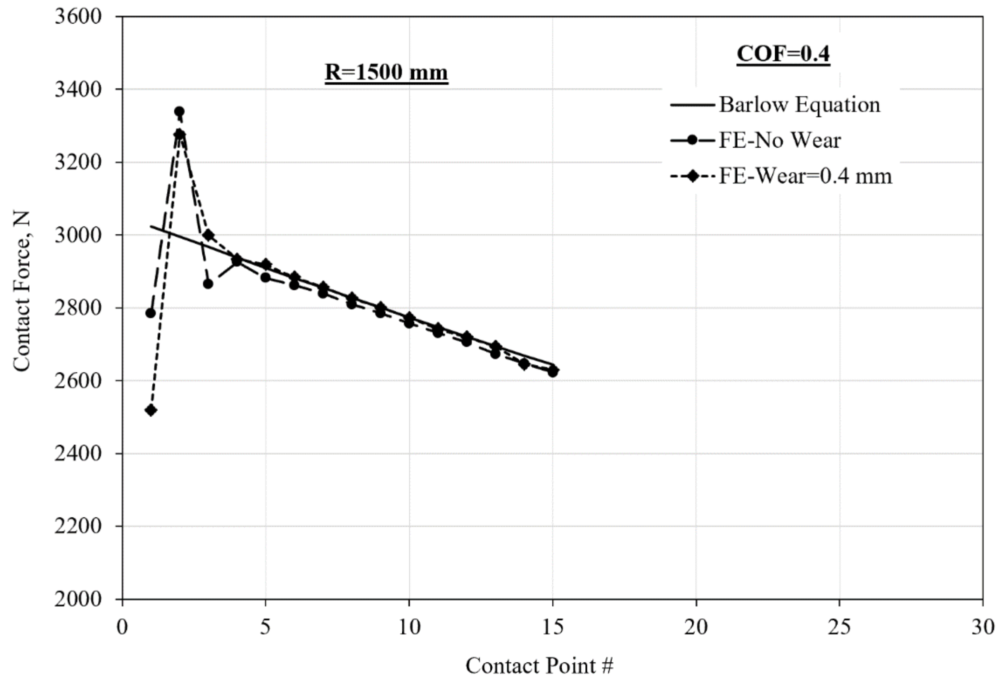


Figure A-5 Contact force results for R = 1500 mm, and COF = 0.4.

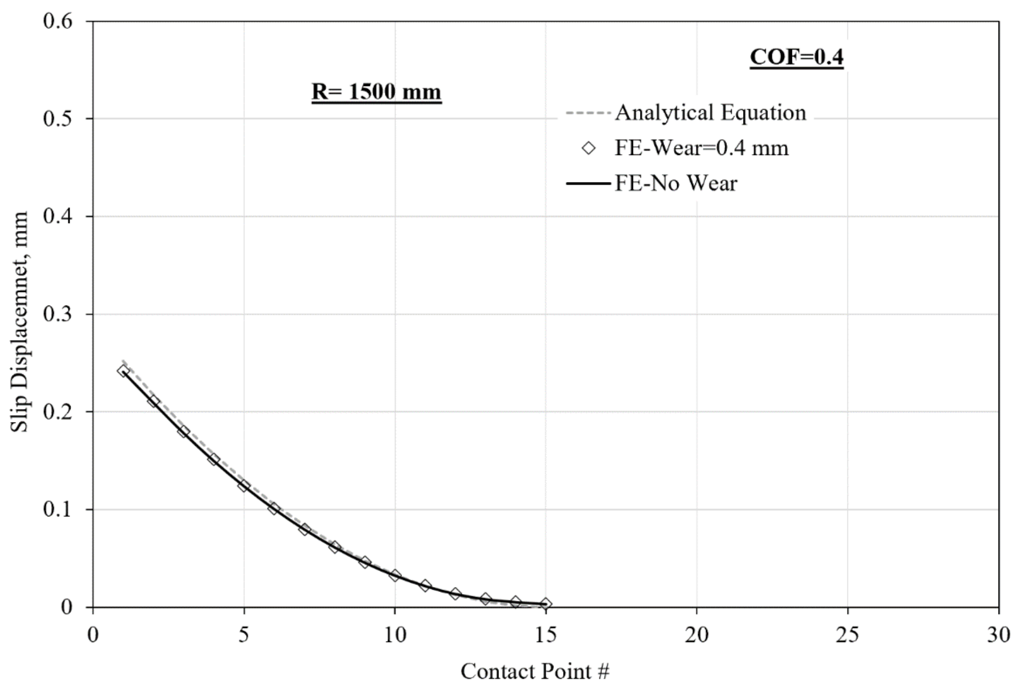


Figure A-6 Slip displacement results for R = 1500 mm, and COF = 0.4.

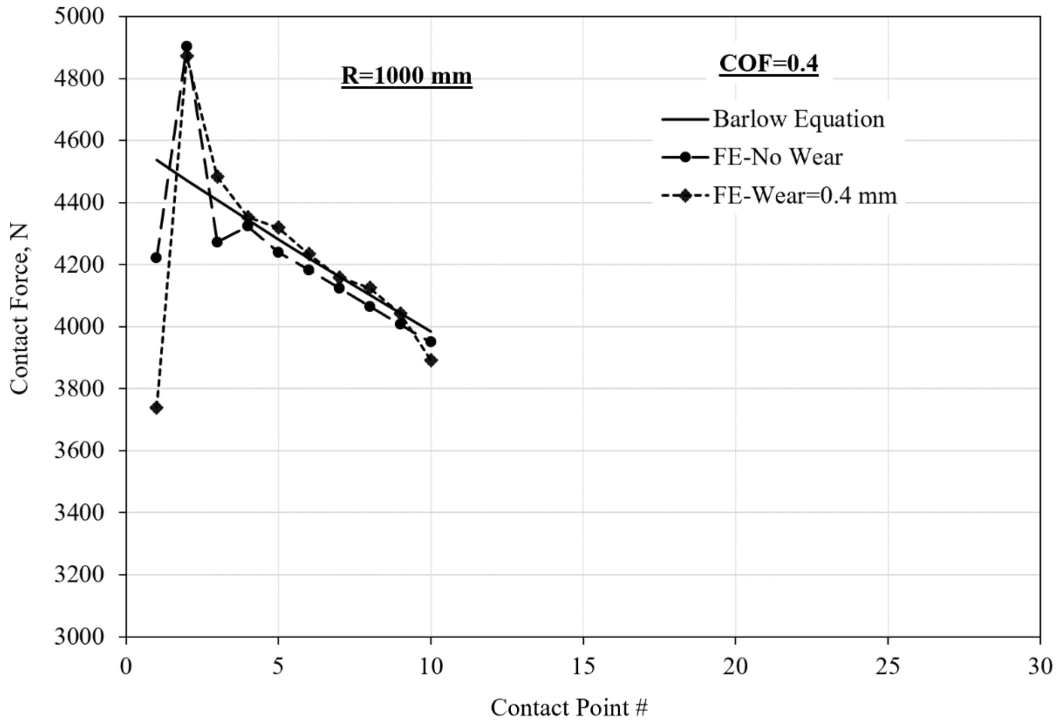


Figure A-7 Contact force results for R = 1000 mm, and COF = 0.4.

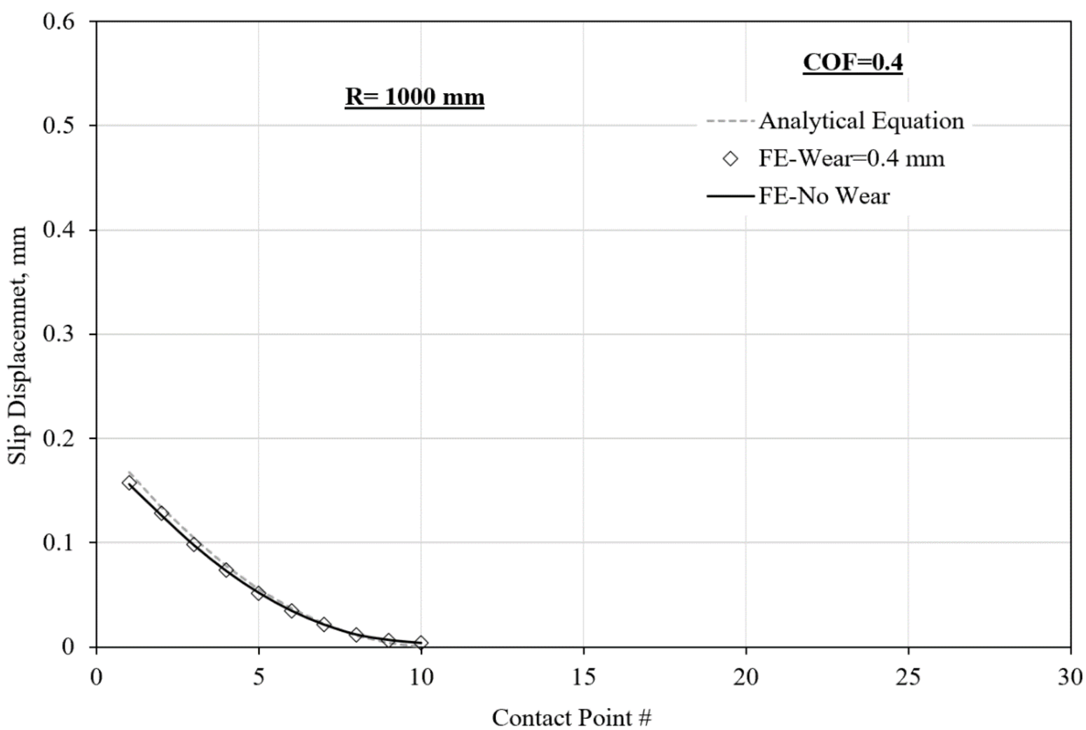


Figure A-8 Slip displacement results for R = 1000 mm, and COF = 0.4.

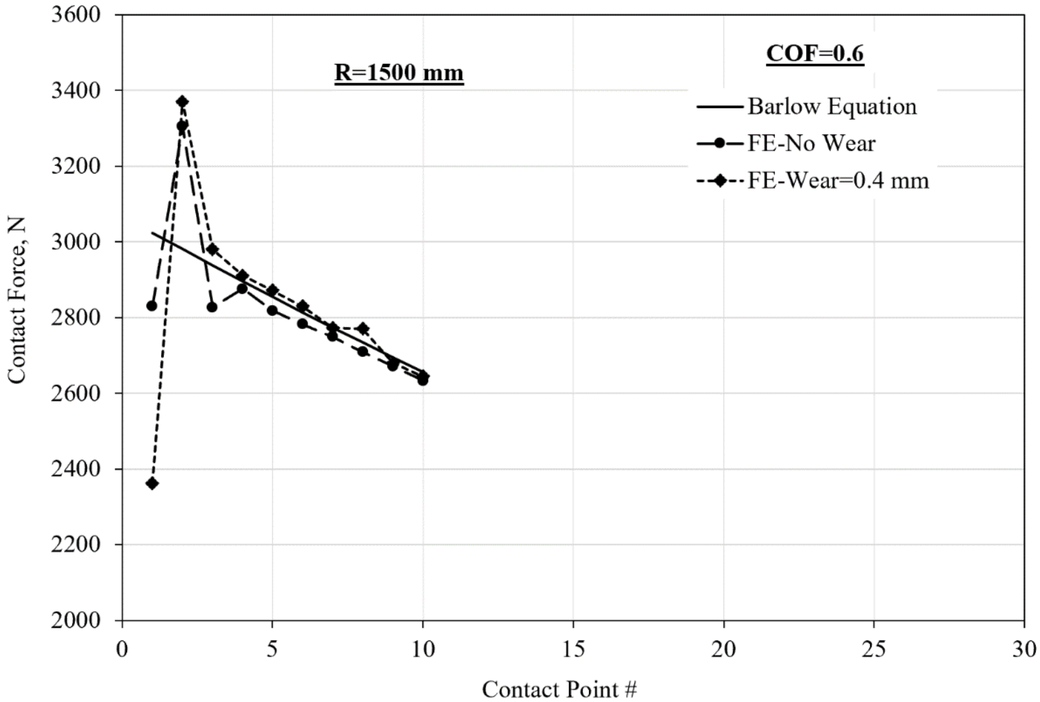


Figure A-9 Contact force results for R = 1500 mm, and COF = 0.6.

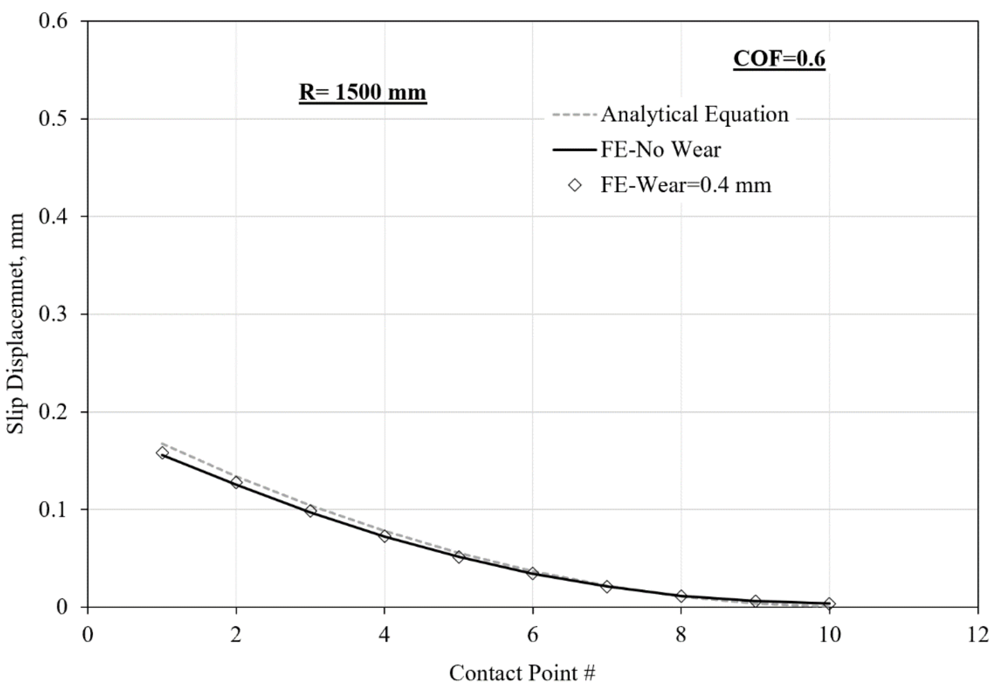


Figure A-10 Slip displacement results for R = 1500 mm, and COF = 0.6.

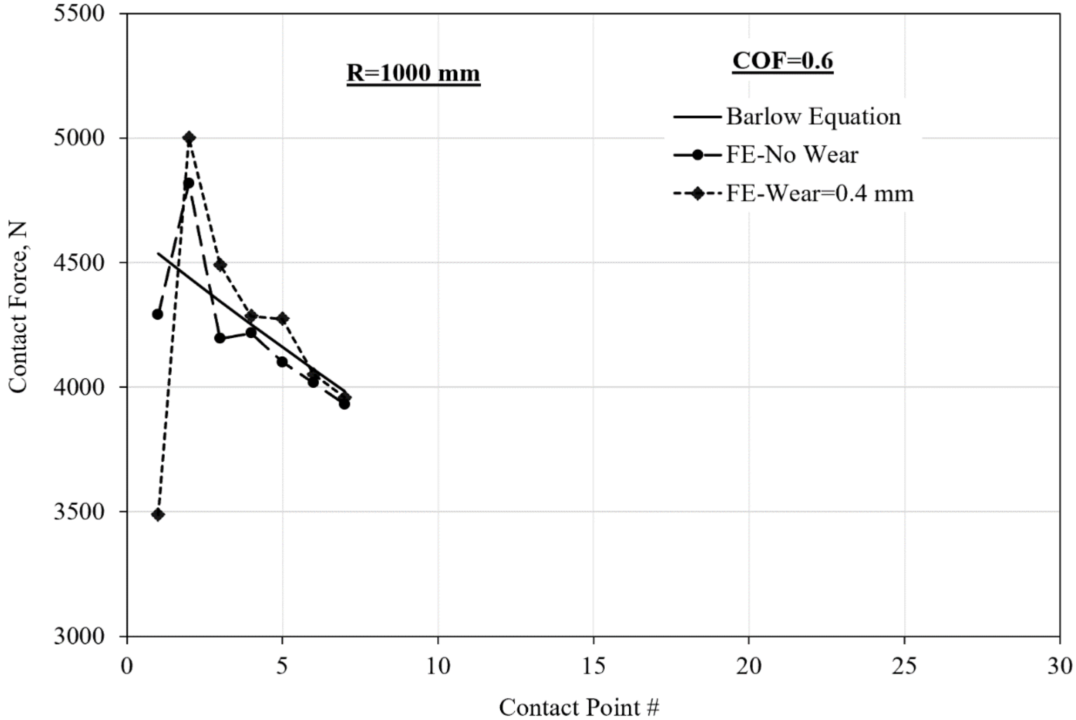


Figure A-11 Contact force results for *R* = 1000 mm, and *COF* = 0.6.

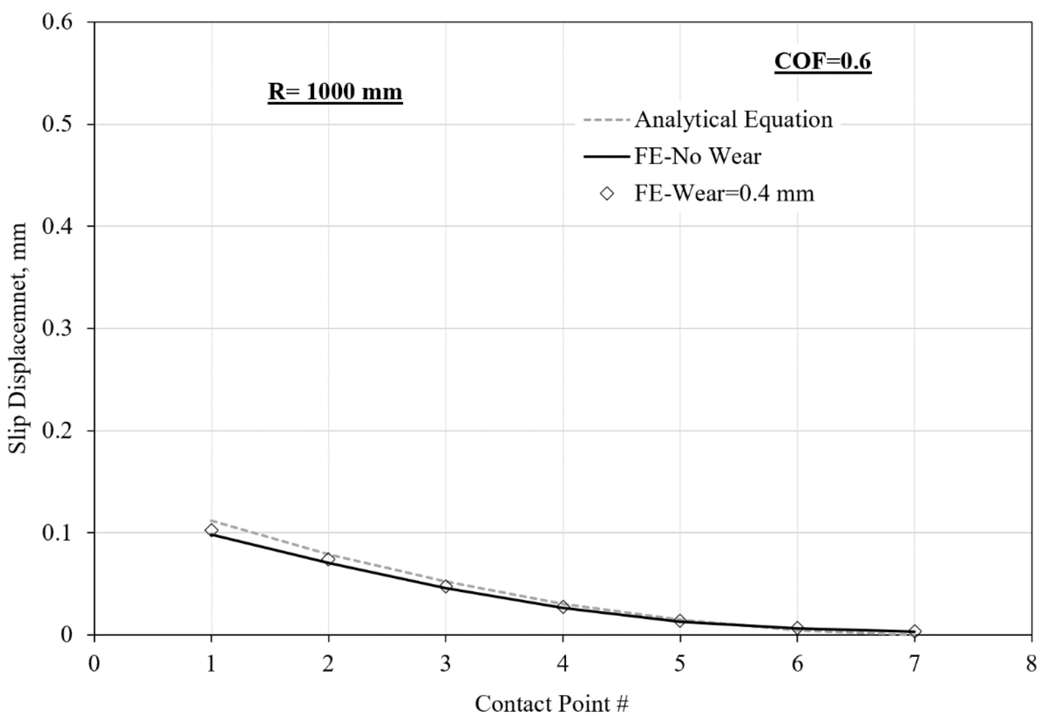


Figure A-12 Slip displacement results for *R* = 1000 mm, and *COF* = 0.6.

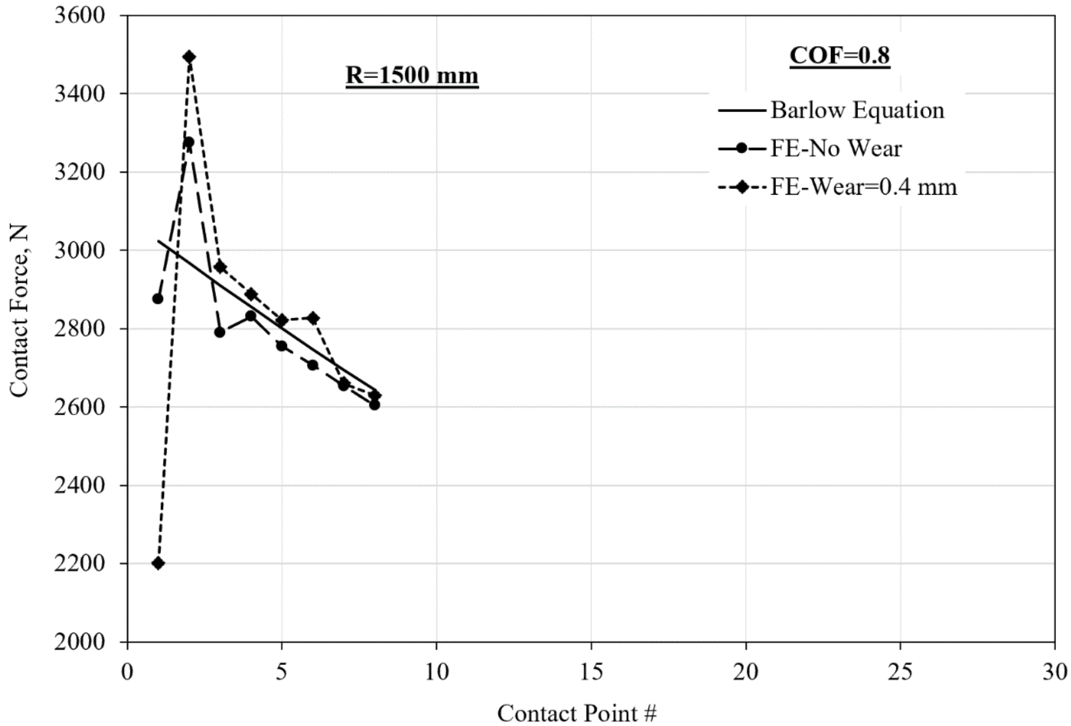


Figure A-13 Contact force results for R = 1500 mm, and COF = 0.8.

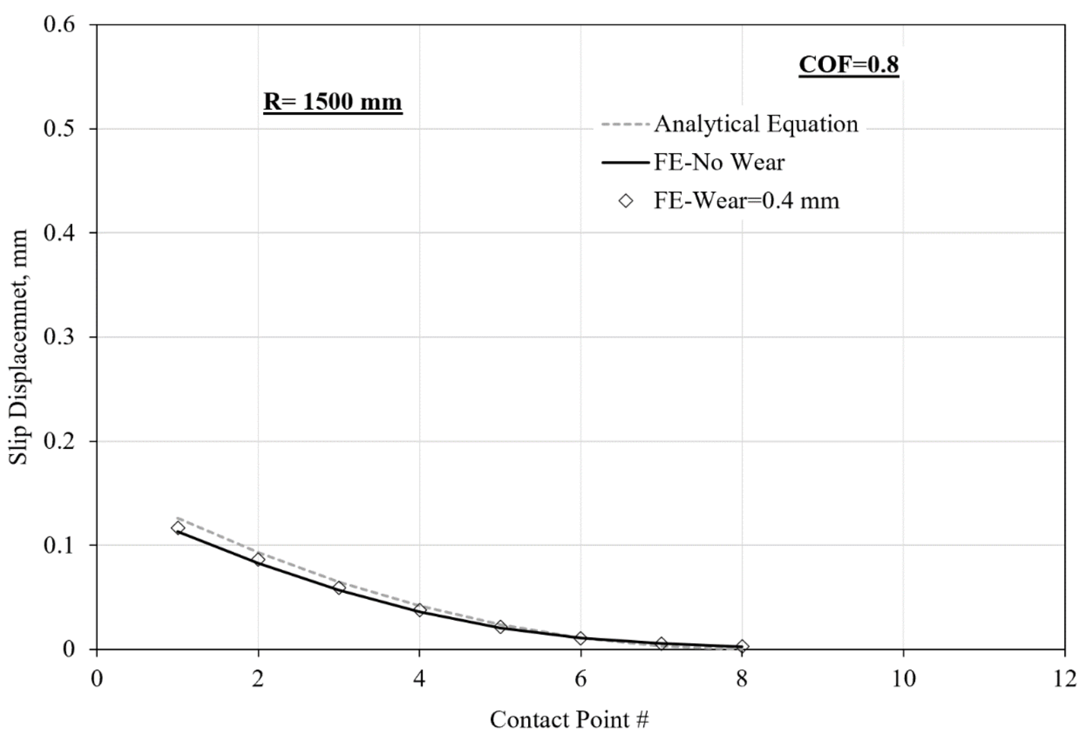


Figure A-14 Slip displacement results for R = 1500 mm, and COF = 0.8.

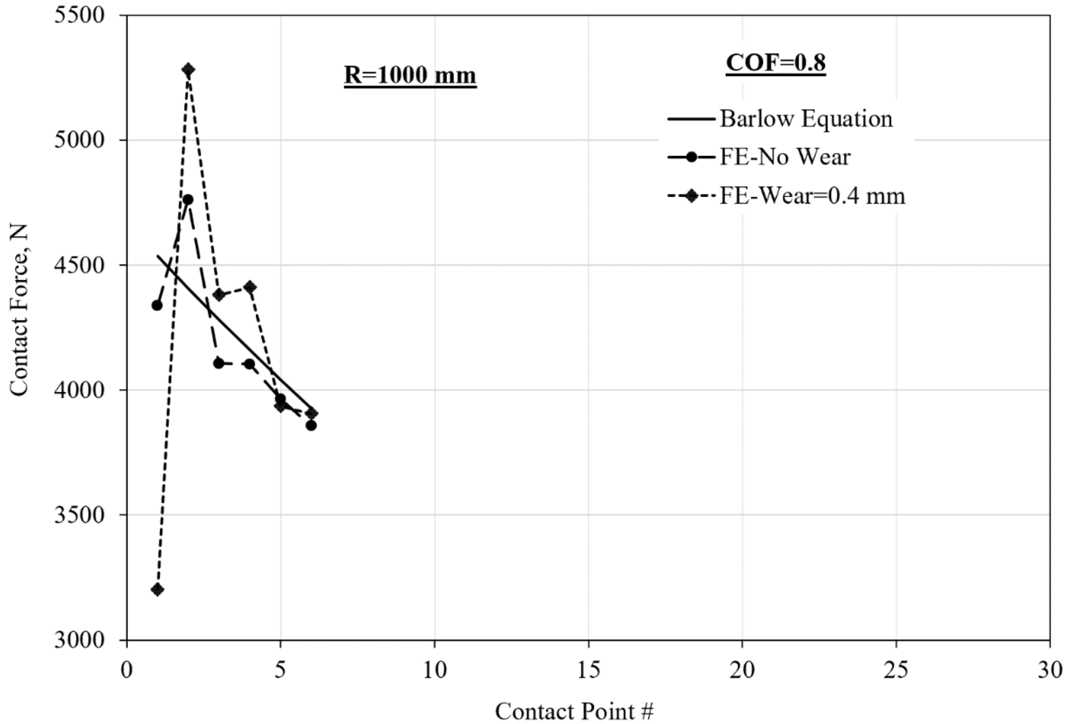


Figure A-15 Contact force results for R = 1000 mm, and COF = 0.8.

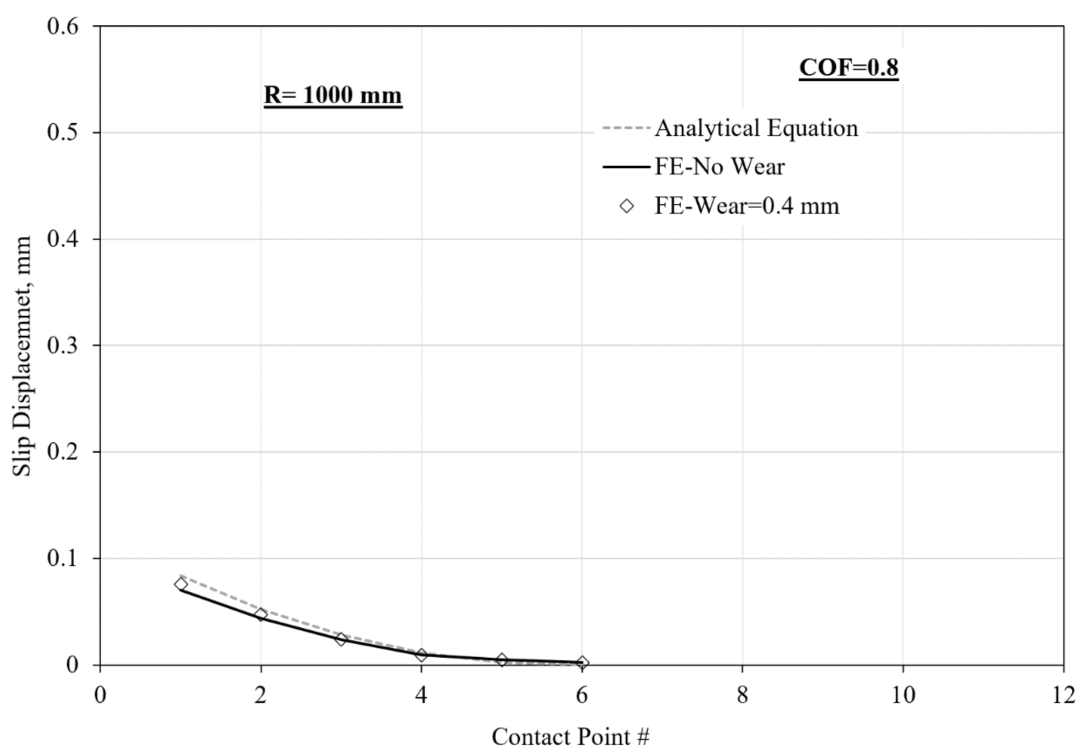


Figure A-16 Slip displacement results for R = 1000 mm, and COF = 0.8.

Appendix B

Table B-1 Input grid for M-DRM analysis for bare wire, R = 500 mm.

Variable /TRIAL	***X1* (*COF*)**	***X2* (*B1*)**	***X3* (*B2*)**	***X4*** (σ'_f)	***X5*** (ε'_f)	SWT	N_f	$Log(N_f)$
1	0.6094	1	1	2675	0.2067	2.887	1378707	6.139
2	0.6462	1	1	2675	0.2067	2.851	1474049	6.169
3	0.7	1	1	2675	0.2067	2.828	1542159	6.188
4	0.7538	1	1	2675	0.2067	2.671	2107833	6.324
5	0.7906	1	1	2675	0.2067	2.811	1594121	6.203
6	0.7	0.5715	1.0000	2675	0.2067	1.710	25652661	7.409
7	0.7	0.7967	1.0000	2675	0.2067	2.367	4101374	6.613
8	0.7	1.0000	1.0000	2675	0.2067	2.828	1542159	6.188
9	0.7	1.2033	1.0000	2675	0.2067	3.314	658108	5.818
10	0.7	1.4285	1.0000	2675	0.2067	3.929	271279	5.433
11	0.7	1.0000	0.7143	2675	0.2067	2.636	2262180	6.355
12	0.7	1.0000	0.8644	2675	0.2067	2.557	2673876	6.427
13	0.7	1.0000	1.0000	2675	0.2067	2.828	1542159	6.188
14	0.7	1.0000	1.1356	2675	0.2067	2.929	1273375	6.105
15	0.7	1.0000	1.2857	2675	0.2067	3.079	973667	5.988
16	0.7	1	1	2316	0.2067	2.828	348715	5.542
17	0.7	1	1	2497	0.2067	2.828	744854	5.872
18	0.7	1	1	2672	0.2067	2.828	1521659	6.182
19	0.7	1	1	2859	0.2067	2.828	3175169	6.502
20	0.7	1	1	3082	0.2067	2.828	7302548	6.863
21	0.7	1	1	2675	0.1296	2.828	1452244	6.162
22	0.7	1	1	2675	0.1645	2.828	1492970	6.174
23	0.7	1	1	2675	0.2041	2.828	1539129	6.187
24	0.7	1	1	2675	0.2532	2.828	1596447	6.203
25	0.7	1	1	2675	0.3215	2.828	1676273	6.224

Table B-2 Input grid for M-DRM analysis for galvanized wire, R = 500 mm.

Variable /Trial	*X1 (COF)*	*X2 (b1)*	*X3 (b2)*	*X4* (σ_f')	*X5* (ε_f')	*SWT*	N_f	$Log(N_f)$
1	0.6094	1	1	2183	1.99	2.887	4775521	6.679
2	0.6462	1	1	2183	1.99	2.851	5241761	6.719
3	0.7000	1	1	2183	1.99	2.828	5581802	6.747
4	0.7538	1	1	2183	1.99	2.671	8608559	6.935
5	0.7906	1	1	2183	1.99	2.811	5845011	6.767
6	0.7	0.5715	1	2183	1.99	1.710	254886219	8.406
7	0.7	0.7967	1	2183	1.99	2.367	21482747	7.332
8	0.7	1.0000	1	2183	1.99	2.828	5581802	6.747
9	0.7	1.2033	1	2183	1.99	3.314	1689323	6.228
10	0.7	1.4285	1	2183	1.99	3.929	476311	5.678
11	0.7	1	0.7143	2183	1.99	2.636	9491241	6.977
12	0.7	1	0.8644	2183	1.99	2.557	11950823	7.077
13	0.7	1	1.0000	2183	1.99	2.828	5581802	6.747
14	0.7	1	1.1356	2183	1.99	2.929	4274277	6.631
15	0.7	1	1.2857	2183	1.99	3.079	2935267	6.468
16	0.7	1	1	1890	1.99	2.828	647422	5.811
17	0.7	1	1	2037	1.99	2.828	1973874	6.295
18	0.7	1	1	2180	1.99	2.828	5477414	6.739
19	0.7	1	1	2333	1.99	2.828	15288474	7.184
20	0.7	1	1	2515	1.99	2.828	47790698	7.679
21	0.7	1	1	2183	1.2477	2.828	5565279	6.745
22	0.7	1	1	2183	1.5840	2.828	5572769	6.746
23	0.7	1	1	2183	1.9650	2.828	5581246	6.747
24	0.7	1	1	2183	2.4376	2.828	5591751	6.748
25	0.7	1	1	2183	3.0948	2.828	5606337	6.749

Table B-3 M-DRM parameters for bare wire, R = 500 mm.

Moment	Entropy	i	0	1	2	3	4
m = 1	2.1003	λ_i	2.023944	4.9213E-12			
		α_i		12.6106			
m = 2	0.5084	λ_i	361.1723	-268.3655	60.6507		
		α_i		0.6005	1.0880		
m = 3	0.5018	λ_i	503.1862	80.2356	-280.7974	-140.3209	
		α_i		1.0577	0.2949	0.7730	
m = 4	0.5007	λ_i	583.6532	3.0958	-234.7744	-63.8870	-168.5974
		α_i		1.8647	-0.0010	0.6399	0.1864

Table B-4 M-DRM parameters for galvanized wire, R = 500 mm.

Moment	Entropy	i	0	1	2	3	4
m = 1	2.2168	λ_i	2.121183	1.3942E-11			
		α_i		11.5442			
m = 2	0.8436	λ_i	318.3254	-229.3842	21.2960		
		α_i		0.4055	1.1167		
m = 3	0.8346	λ_i	510.5722	86.1741	-363.5205	-110.3246	
		α_i		0.9169	0.2213	0.7384	
m = 4	0.8344	λ_i	542.3963	-29.5924	64.1795	-440.3457	31.3320
		α_i		1.3812	-8.7018	0.1604	1.4179

Table B-5 Sensitivity index results for bare wire, R = 500 mm.

Variable	Parameter	S_i	S_{Ti}	$S_{Ti} - S_i$
X1	COF-Coefficient of friction	0.0245	0.0247	0.0002
X2	b_1-Bias factor for contact force	0.5560	0.5577	0.0017
X3	b_2-Bias factor for slip displacement	0.0816	0.0821	0.0005
X4	σ_f'-Fatigue strength coefficient	0.3359	0.3375	0.0016
X5	ε_f'-Fatigue ductility factor	0.0000	0.0000	0.0000
Sum		0.9980	1.0020	

Table B-6 Sensitivity index results for galvanized wire, R = 500 mm.

Variable	Parameter	S_i	S_{Ti}	$S_{Ti} - S_i$
X1	COF-Coefficient of friction	0.0244	0.0245	0.0001
X2	b_1-Bias factor for contact force	0.5634	0.5645	0.0011
X3	b_2 -Bias factor for slip displacement	0.0829	0.0832	0.0003
X4	σ_f'-Fatigue strength coefficient	0.3273	0.3283	0.0009
X5	ε_f'-Fatigue ductility factor	0.0007	0.0007	0.0000
Sum		0.9988	1.0012	

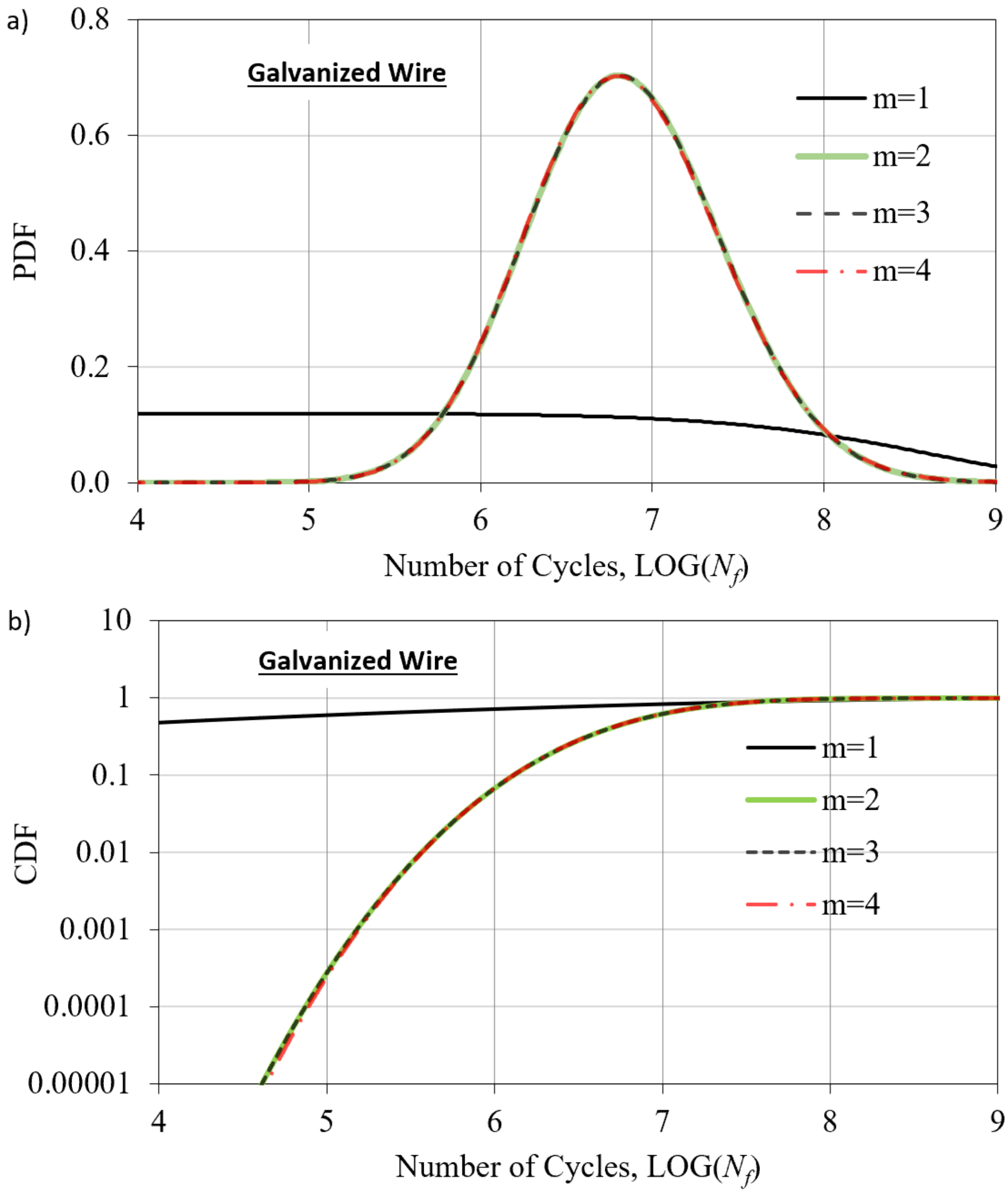


Figure B-1 Comparing M-DRM results with different numbers of terms for galvanized wires, R = 500 mm.

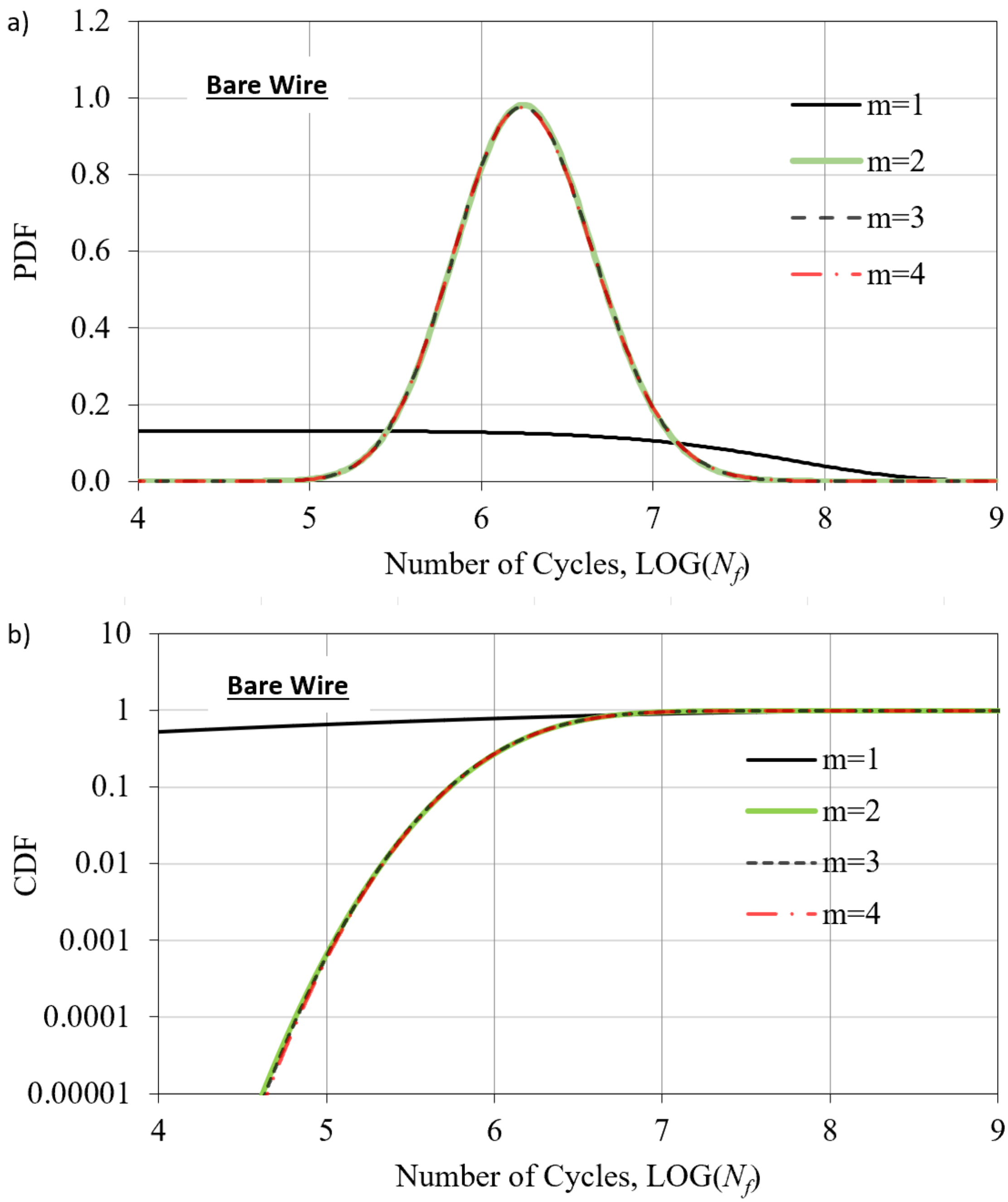


Figure B-2 Comparing M-DRM results with different numbers of terms for bare wires, R = 500 mm.

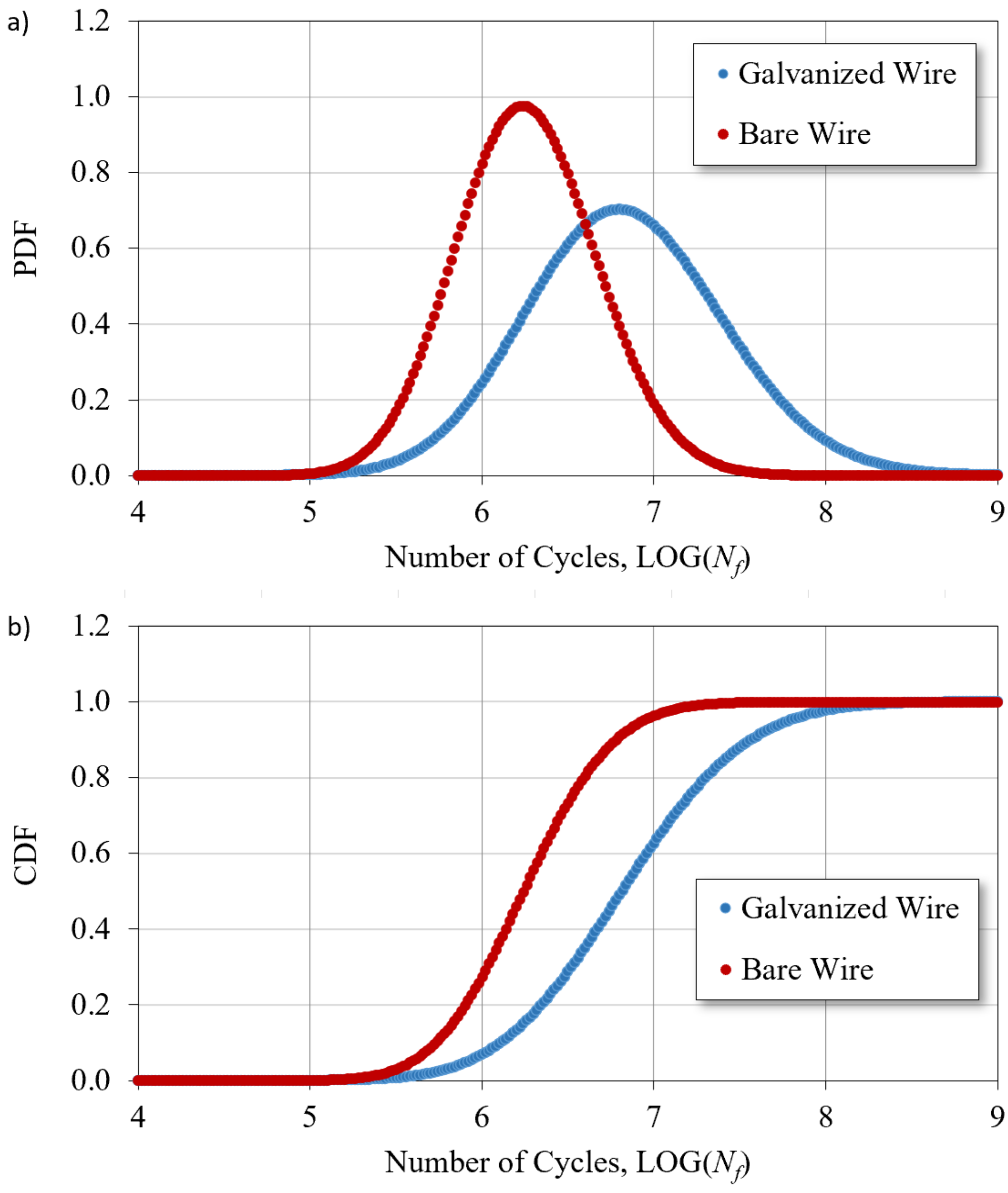


Figure B-3 PDF and CDF results using M-DRM for bare and galvanized wires, R = 500 mm.

Table B-7 Input grid for M-DRM analysis for bare wire, *R* = 1500 mm.

Variable /Trial	*X1 (COF)*	*X2 (b1)*	*X3 (b2)*	*X4* (σ'_f)	*X5* (ε'_f)	*SWT*	N_f	$Log(N_f)$
1	0.6094	1	1	2675	0.2067	3.2229	762650	5.8823
2	0.6462	1	1	2675	0.2067	3.1796	819706	5.9137
3	0.7000	1	1	2675	0.2067	3.4130	563000	5.7505
4	0.7538	1	1	2675	0.2067	3.3687	603174	5.7804
5	0.7906	1	1	2675	0.2067	3.4296	548816	5.7394
6	0.7	0.5715	1	2675	0.2067	2.7944	1645292	6.2162
7	0.7	0.7967	1	2675	0.2067	2.8025	1619655	6.2094
8	0.7	1.0000	1	2675	0.2067	3.4130	563000	5.7505
9	0.7	1.2033	1	2675	0.2067	3.9068	279244	5.4460
10	0.7	1.4285	1	2675	0.2067	4.4763	141036	5.1493
11	0.7	1	0.714303	2675	0.2067	3.0636	1000244	6.0001
12	0.7	1	0.864437	2675	0.2067	3.2465	733617	5.8655
13	0.7	1	1	2675	0.2067	3.4130	563000	5.7505
14	0.7	1	1.135563	2675	0.2067	3.5710	444117	5.6475
15	0.7	1	1.285697	2675	0.2067	3.7365	351047	5.5454
16	0.7	1	1	2316.2319	0.2067	3.4130	137458	5.1382
17	0.7	1	1	2496.6800	0.2067	3.4130	280945	5.4486
18	0.7	1	1	2671.6627	0.2067	3.4130	555807	5.7449
19	0.7	1	1	2858.9093	0.2067	3.4130	1131203	6.0535
20	0.7	1	1	3081.6351	0.2067	3.4130	2549302	6.4064
21	0.7	1	1	2675	0.1296	3.4130	515307	5.7121
22	0.7	1	1	2675	0.1645	3.4130	536897	5.7299
23	0.7	1	1	2675	0.2041	3.4130	561392	5.7493
24	0.7	1	1	2675	0.2532	3.4130	591847	5.7722
25	0.7	1	1	2675	0.3215	3.4130	634344	5.8023

Table B-8 Input grid for M-DRM analysis for galvanized wire, R = 1500 mm.

Variable /Trial	*X1 (COF)*	*X2 (b1)*	*X3 (b2)*	*X4* $(\boldsymbol{\sigma'_f})$	*X5* $(\boldsymbol{\varepsilon'_f})$	*SWT*	$\boldsymbol{N_f}$	$\boldsymbol{Log}(\boldsymbol{N_f})$
1	0.6094	1	1	2183	1.99	3.2229	2080937	6.3183
2	0.6462	1	1	2183	1.99	3.1796	2303902	6.3625
3	0.7000	1	1	2183	1.99	3.4130	1353753	6.1315
4	0.7538	1	1	2183	1.99	3.3687	1492955	6.1740
5	0.7906	1	1	2183	1.99	3.4296	1305531	6.1158
6	0.7	0.5715	1	2183	1.99	2.7944	6107323	6.7859
7	0.7	0.7967	1	2183	1.99	2.8025	5975521	6.7764
8	0.7	1.0000	1	2183	1.99	3.4130	1353753	6.1315
9	0.7	1.2033	1	2183	1.99	3.9068	496555	5.6960
10	0.7	1.4285	1	2183	1.99	4.4763	185594	5.2686
11	0.7	1	0.7143	2183	1.99	3.0636	3048380	6.4841
12	0.7	1	0.8644	2183	1.99	3.2465	1969924	6.2944
13	0.7	1	1.0000	2183	1.99	3.4130	1353753	6.1315
14	0.7	1	1.1356	2183	1.99	3.5710	965622	5.9848
15	0.7	1	1.2857	2183	1.99	3.7365	689734	5.8387
16	0.7	1	1	1890.2184	1.99	3.4130	167329	5.2236
17	0.7	1	1	2037.4774	1.99	3.4130	487818	5.6883
18	0.7	1	1	2180.2764	1.99	3.4130	1328735	6.1234
19	0.7	1	1	2333.0836	1.99	3.4130	3677537	6.5656
20	0.7	1	1	2514.8444	1.99	3.4130	11449918	7.0588
21	0.7	1	1	2183	1.2477	3.4130	1342393	6.1279
22	0.7	1	1	2183	1.5840	3.4130	1347548	6.1295
23	0.7	1	1	2183	1.9650	3.4130	1353372	6.1314
24	0.7	1	1	2183	2.4376	3.4130	1360576	6.1337
25	0.7	1	1	2183	3.0948	3.4130	1370556	6.1369

Table B-9 M-DRM parameters for bare wire, R = 1500 mm.

Moments	Entropy	i	0	1	2	3	4
m=1	2.0321	λ_i	1.954423	2.1414E-11			
		α_i		12.3075			
m=2	0.4080	λ_i	387.4132	-255.4907	15.8790		
		α_i		0.4620	1.4062		
m=3	0.4016	λ_i	515.5412	80.1315	-271.6791	-151.3373	
		α_i		1.0216	0.4183	0.5962	
m=4	0.3976	λ_i	692.9105	2.4973	-367.9469	-30.6709	-213.3714
		α_i		2.1327	0.0109	1.0785	0.0167

Table B-10 M-DRM parameters for galvanized wire, R = 1500 mm.

Moments	Entropy	i	0	1	2	3	4
m=1	2.1203	λ_i	2.047372	9.4191E-12			
		α_i		12.1503			
m=2	0.7669	λ_i	362.7021	-275.3934	15.3736		
		α_i		0.3237	1.1896		
m=3	0.7659	λ_i	386.5421	61.8517	-234.8427	-111.9732	
		α_i		1.0038	0.2054	0.7385	
m=4	0.7620	λ_i	571.6363	1.7702	-299.1975	-29.4097	-194.6583
		α_i		1.9807	-0.0117	0.8977	-0.0051

Table B-11 Sensitivity index results for bare wire, R = 1500 mm.

Variable	Parameter	S_i	S_{Ti}	$S_{Ti} - S_i$
X1	COF-Coefficient of friction	0.035462	0.035599	0.000138
X2	b_1-Bias factor for contact force	0.539976	0.540975	0.000999
X3	b_2-Bias factor for slip displacement	0.048384	0.048569	0.000186
X4	σ_f'-Fatigue strength coefficient	0.373193	0.374135	0.000942
X5	ε_f'-Fatigue ductility factor	0.00185	0.001858	7.44E-06
Sum		0.998864	1.001136	

Table B-12 Sensitivity index results for galvanized wire, R = 1500 mm.

Variable	Parameter	S_i	S_{Ti}	$S_{Ti} - S_i$
X1	COF-Coefficient of friction	0.035183612	0.035429	0.000245
X2	b_1-Bias factor for contact force	0.52931437	0.531106	0.001792
X3	b_2 -Bias factor for slip displacement	0.047912311	0.048242	0.000329
X4	σ_f'-Fatigue strength coefficient	0.385545583	0.387251	0.001705
X5	ε_f'-Fatigue ductility factor	8.93367E-06	9E-06	6.45E-08
Sum		0.99796481	1.002036	

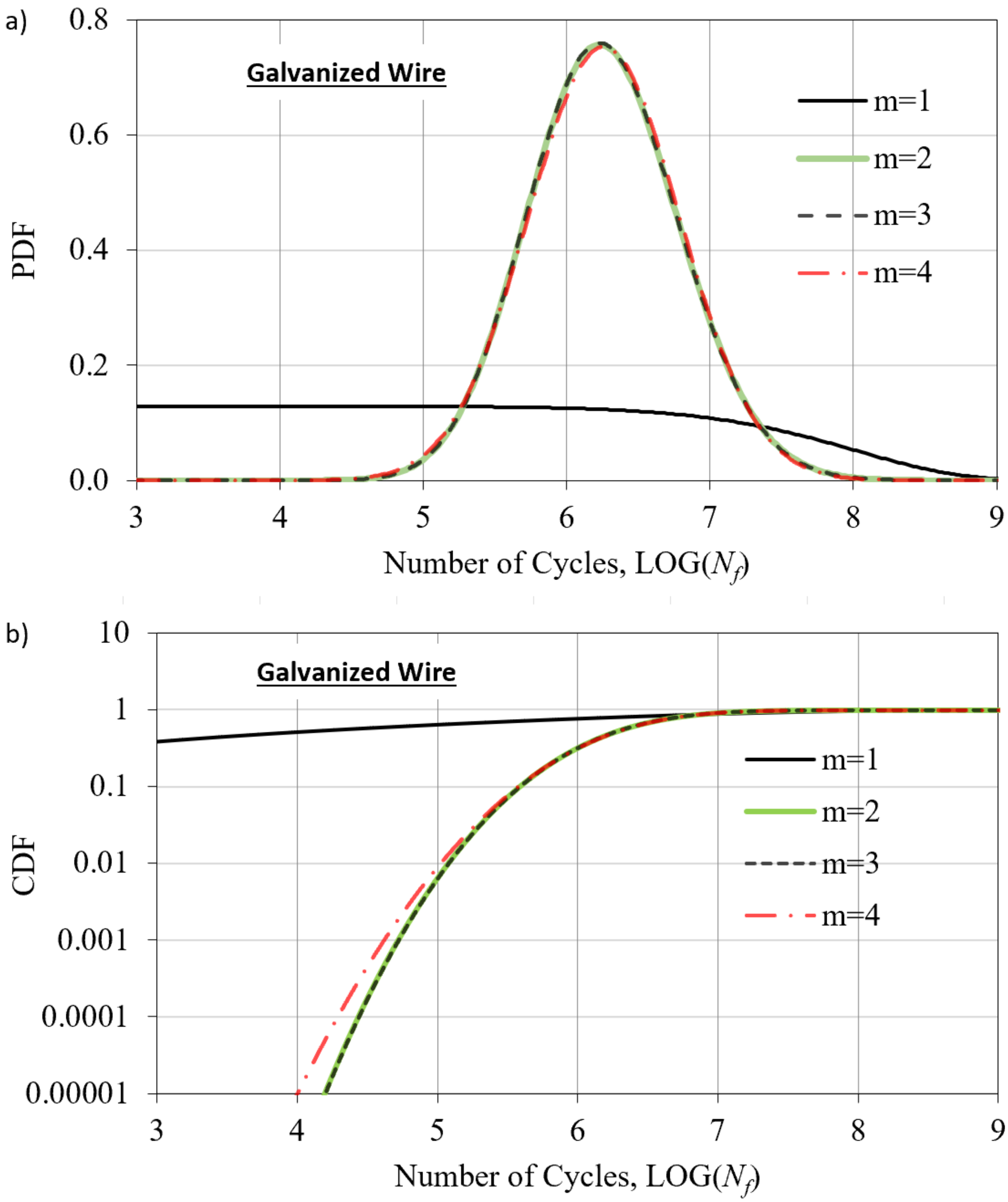


Figure B-4 Comparing M-DRM results with different numbers of terms for galvanized wires, R = 1500 mm.

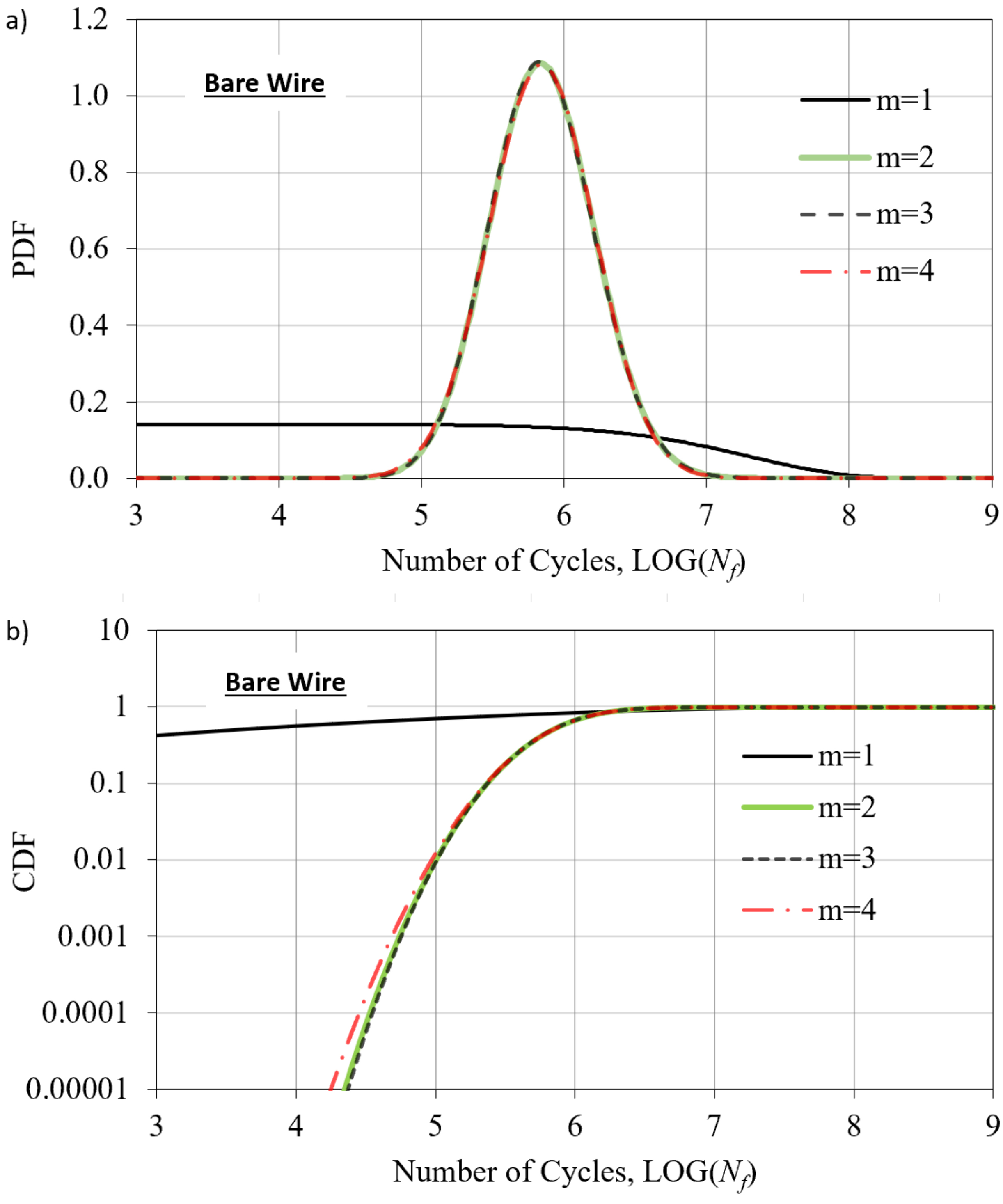


Figure B-5 Comparing M-DRM results with different numbers of terms for bare wires, R = 1500 mm.

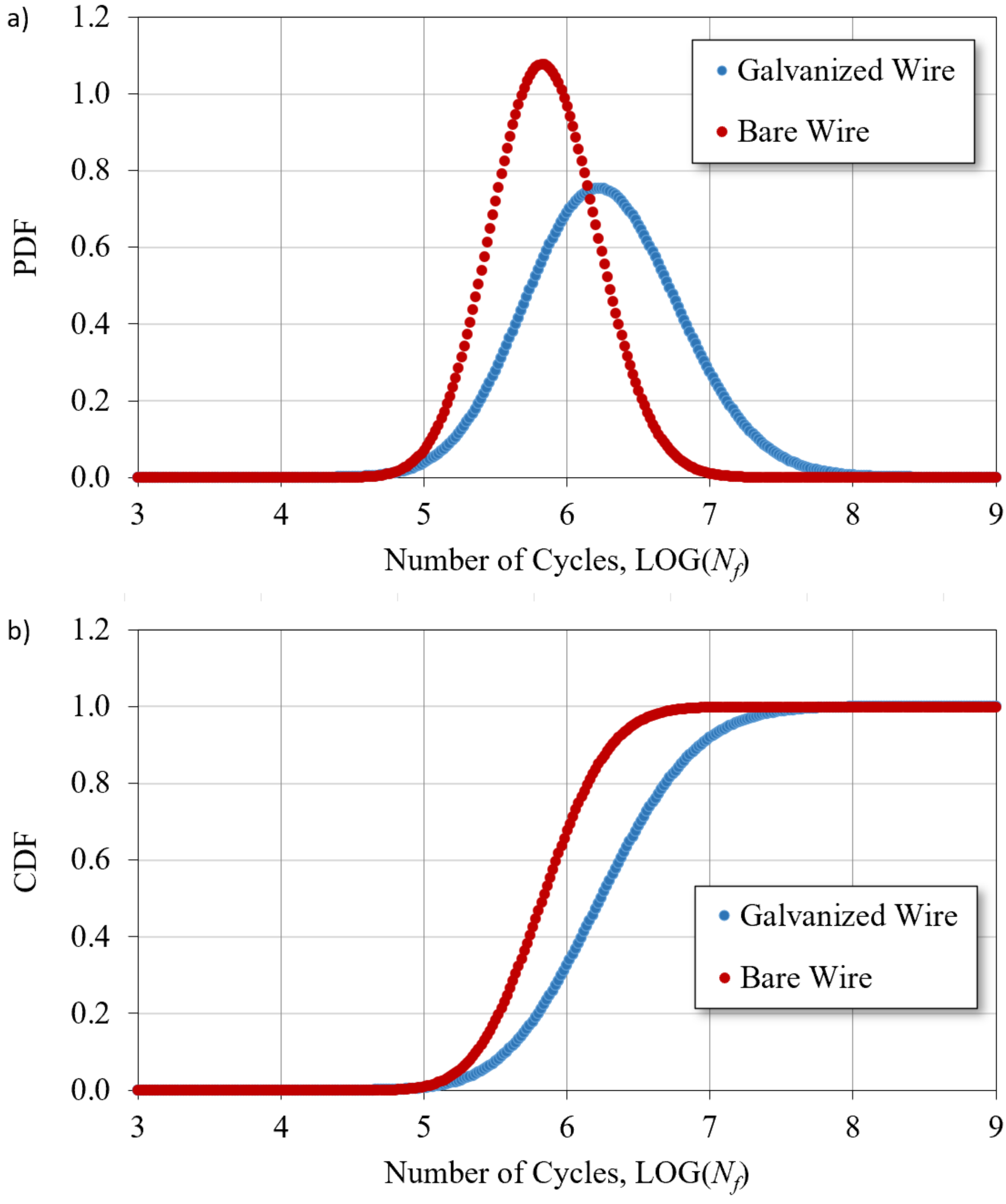


Figure B-6 PDF and CDF results using M-DRM for bare and galvanized wires, R = 1500 mm.

Appendix C

In this appendix, drawings of the fretting fatigue test setup designed for the current project are attached. The parts are marked with letters A to G in the following figure.

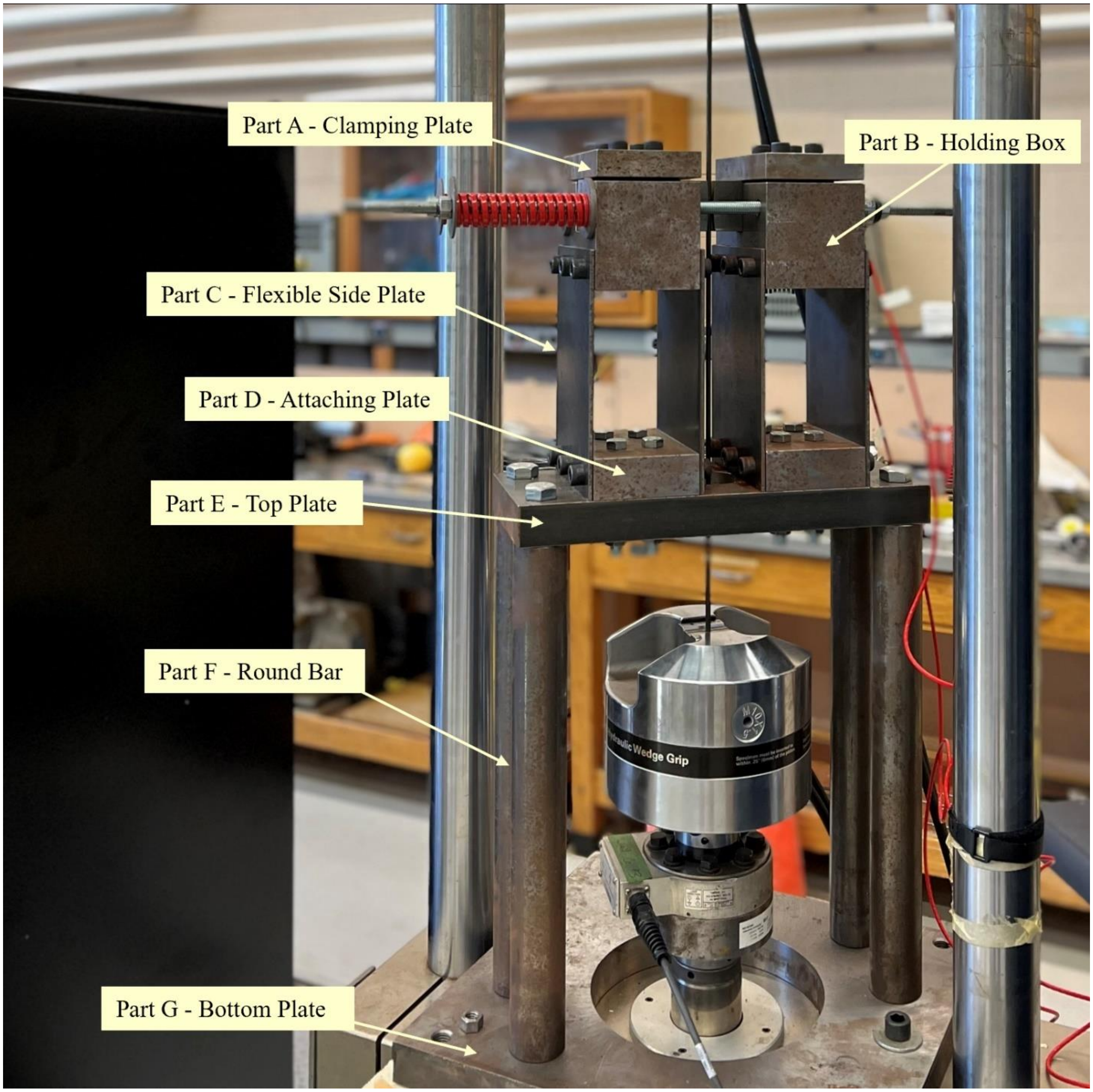


Figure C-1 Fretting fatigue test setup.

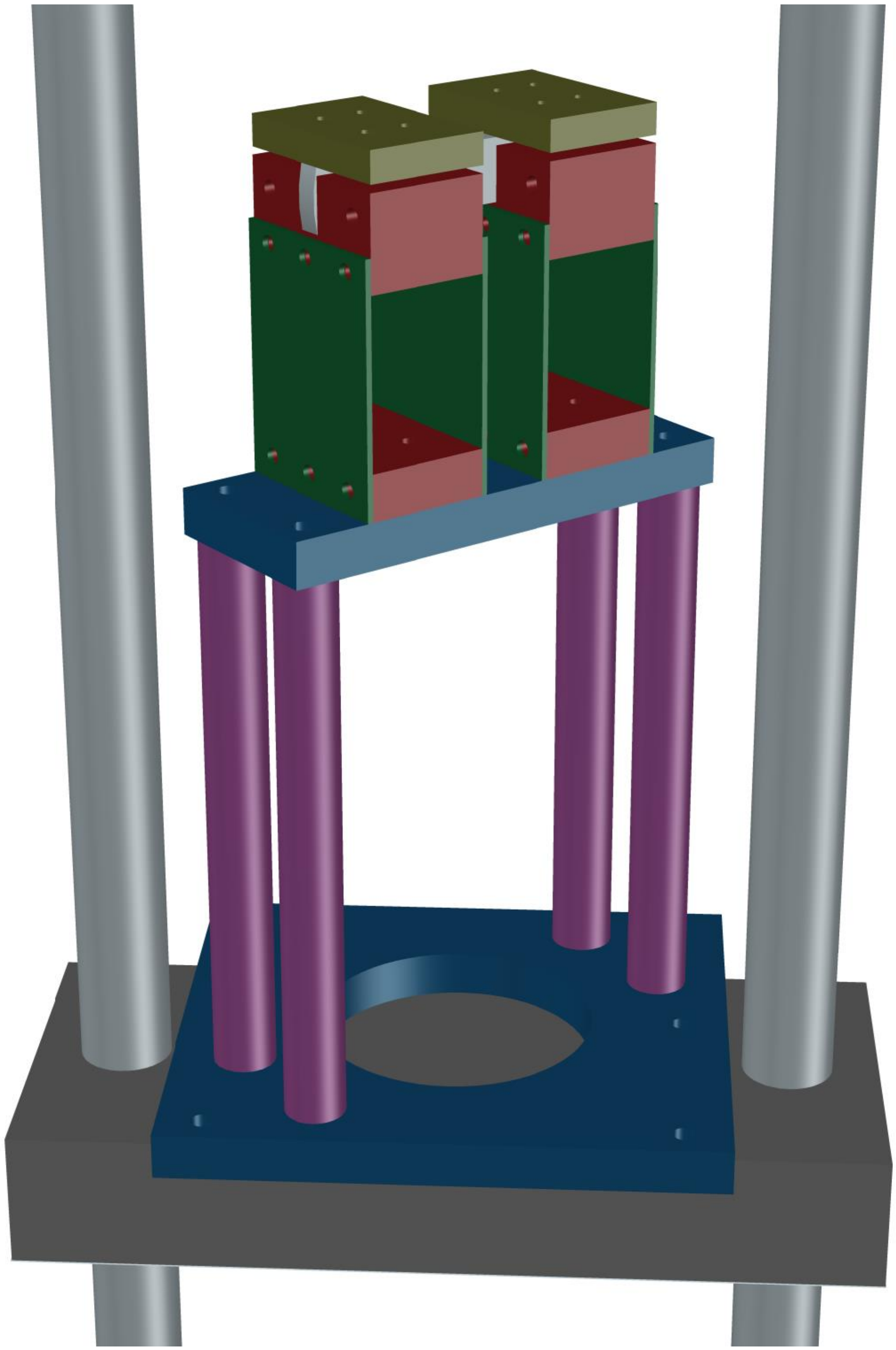


Figure C-2 Schematic front view of the test setup.

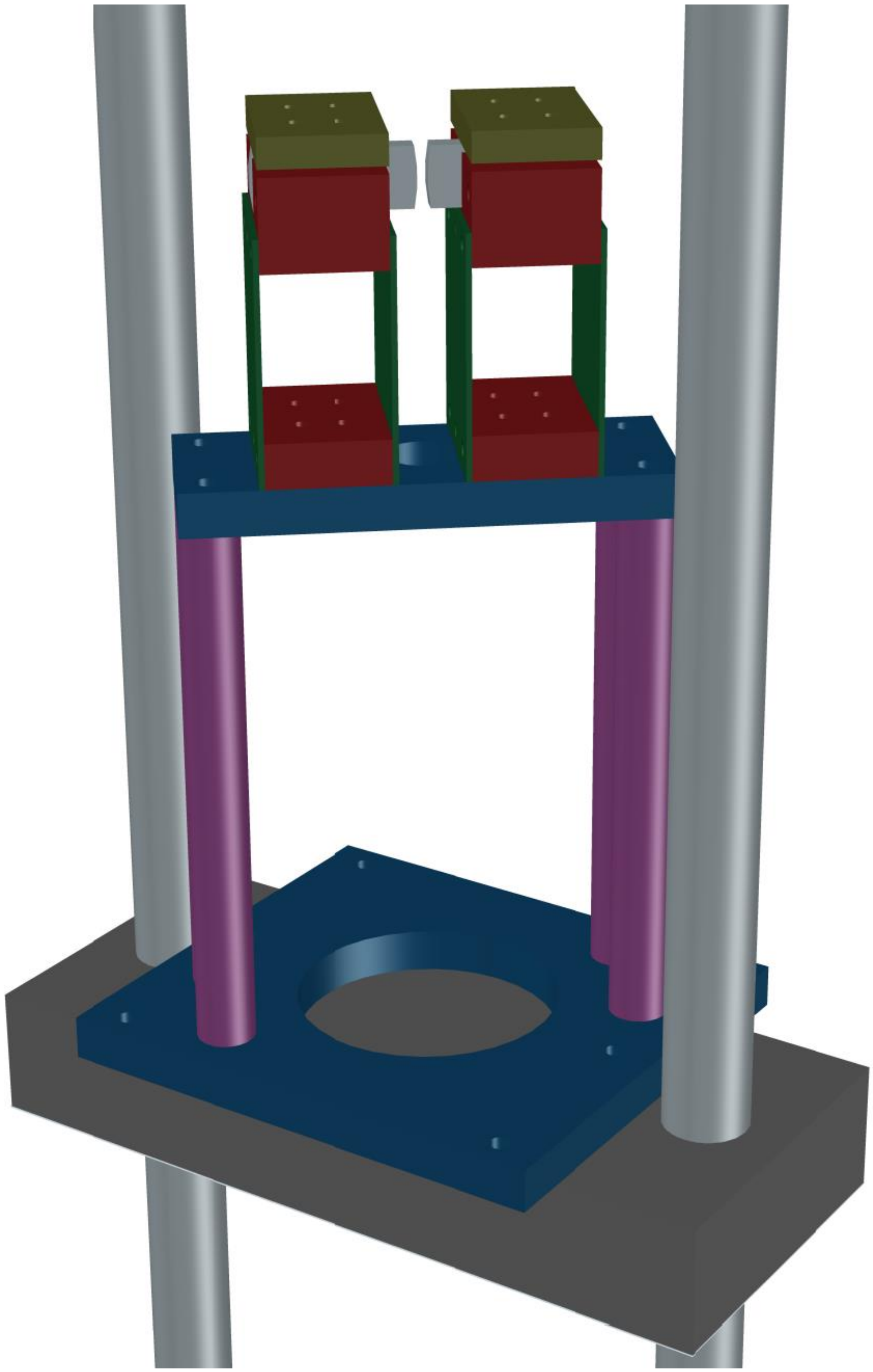


Figure C-3 Schematic side view of the test setup.

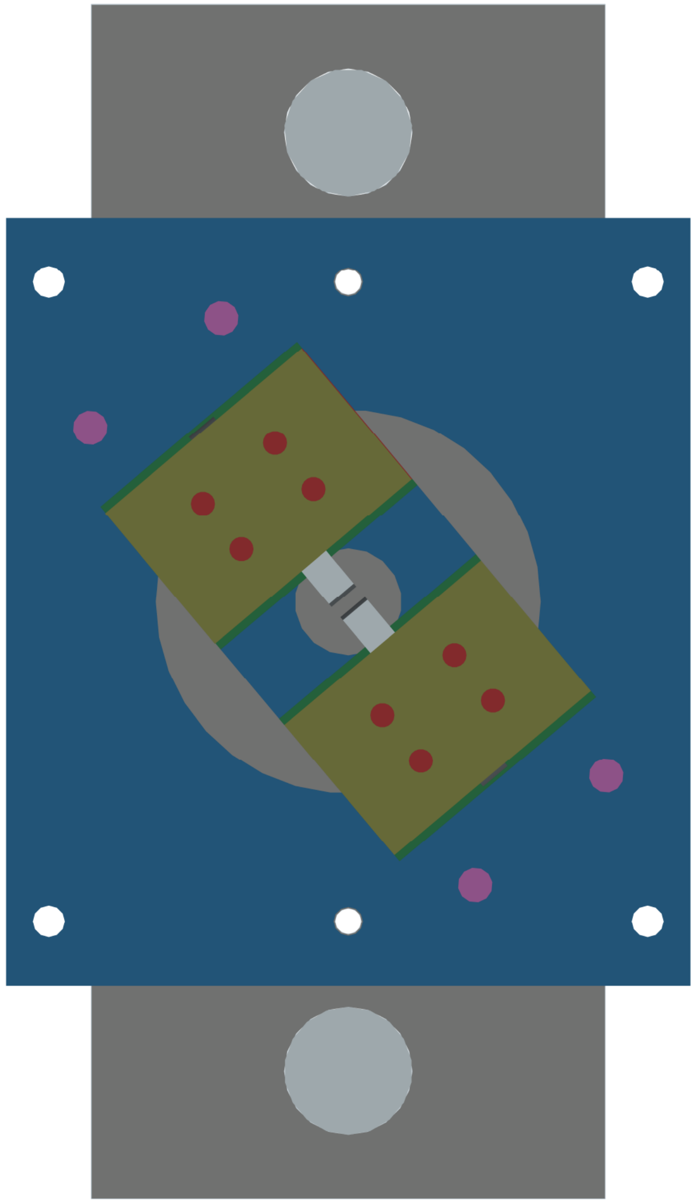
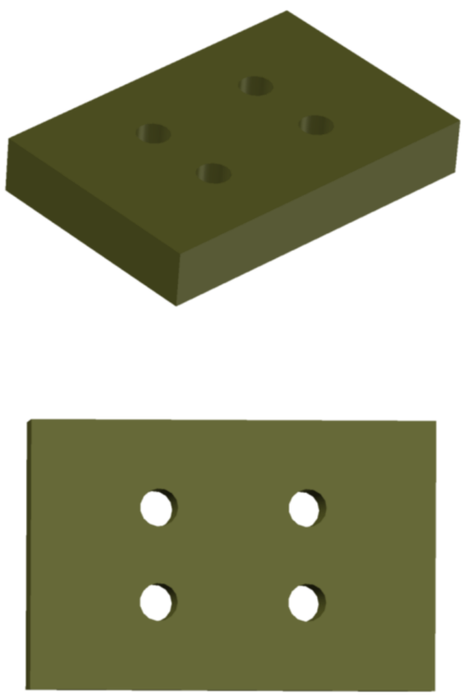


Figure C-4 Schematic top view of the test setup.

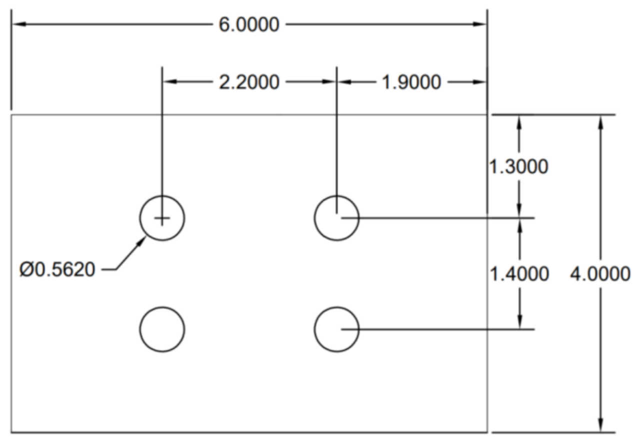
Part A

All dimensions are in inches.
Quantity: 2 pieces.
Material: Cold rolled steel.
Plate 4 in. × 6 in.
Thickness= 1 in.

3D Views



Front and back view



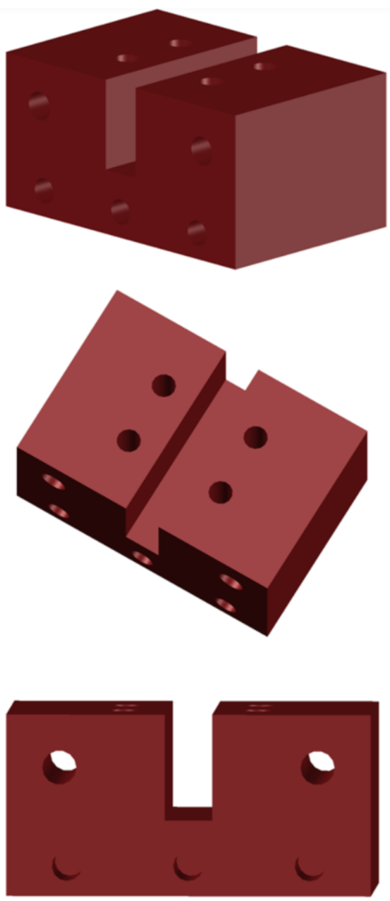
Typical clearance thru holes

Figure C-5 Drawing of Part A.

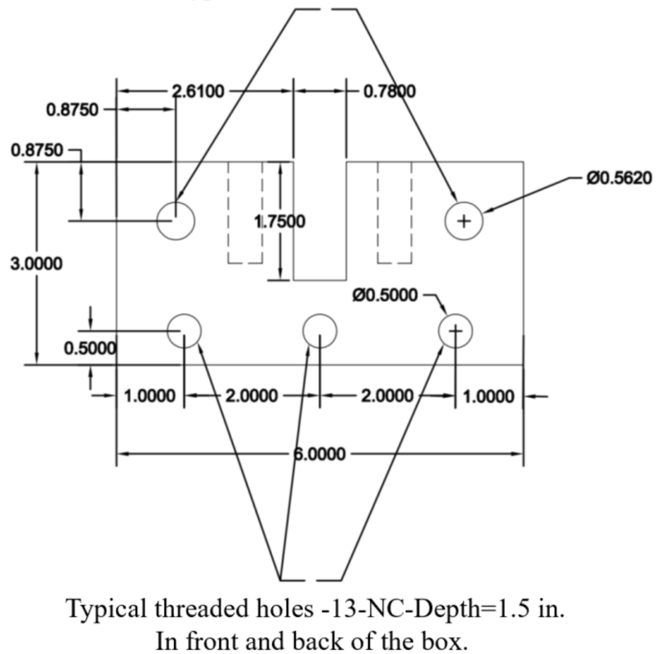
Part B

All dimensions are in inches.
Quantity: 2 pieces.
Material: Cold rolled steel.
Box 6 in. × 4 in. × 3 in.

3D Views



Front and back view



Top View

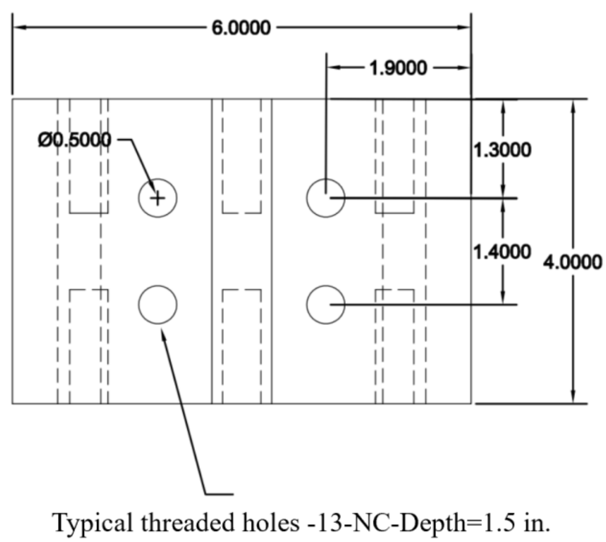
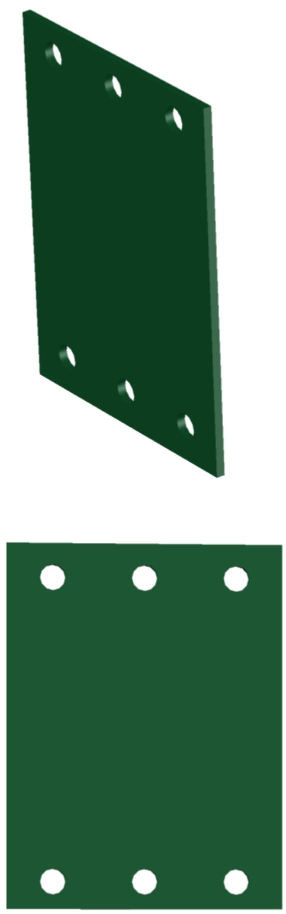


Figure C-6 Drawing of Part B.

Part C

All dimensions are in inches.
Quantity: 4 pieces.
Material: Cold rolled steel.
Plate 8 in. × 6 in.
Thickness= 0.1875 in.

3D Views



Front view

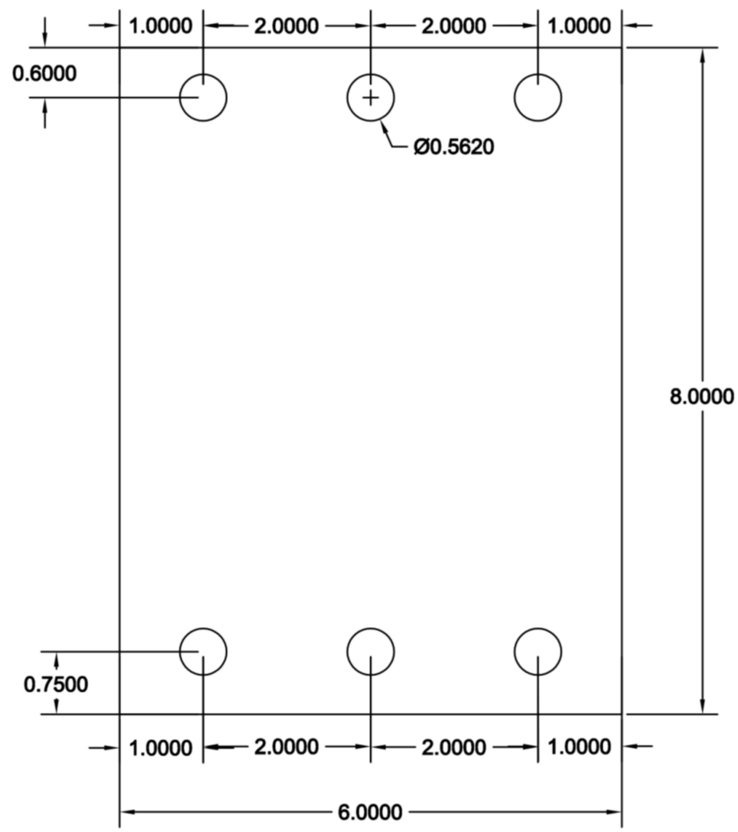
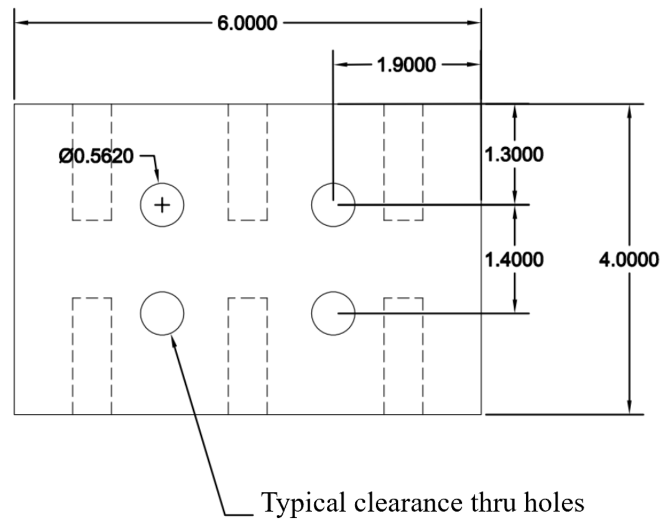


Figure C-7 Drawing of Part C.

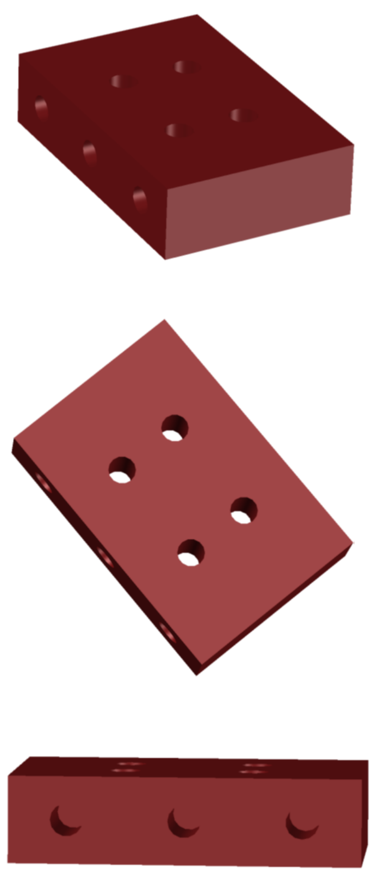
Part D

All dimensions are in inches.
Quantity: 2 pieces.
Material: Cold rolled steel.
Plate 4 in. × 6 in.
Thickness= 1.5 in.

Front view



3D Views



side view

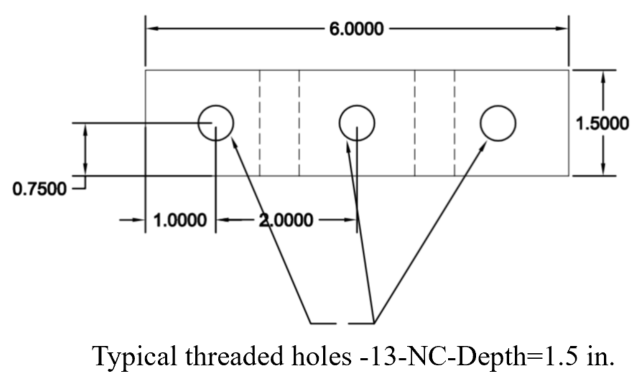
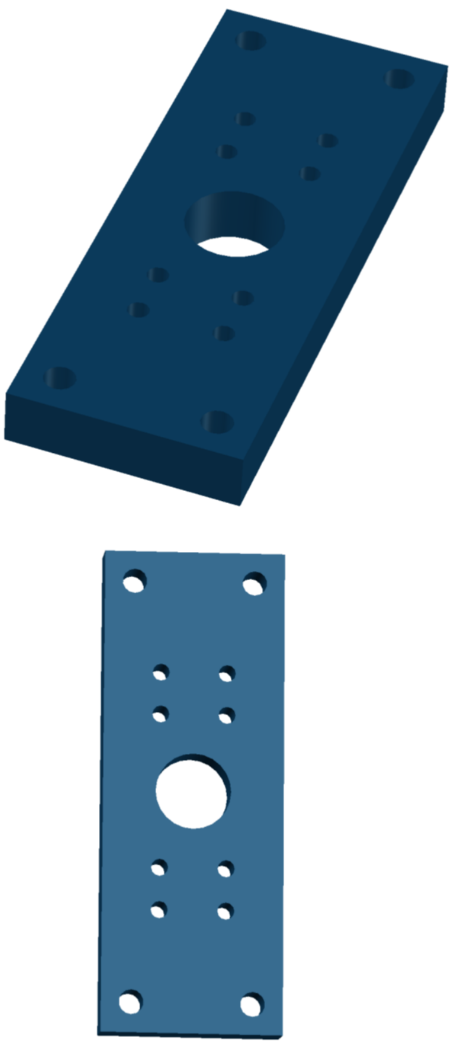


Figure C-8 Drawing of Part D.

Part E

All dimensions are in inches.
Quantity: 1 piece.
Material: Cold rolled steel.
Plate 16 in. × 6 in.
Thickness= 1.5 in.

3D Views



Front view

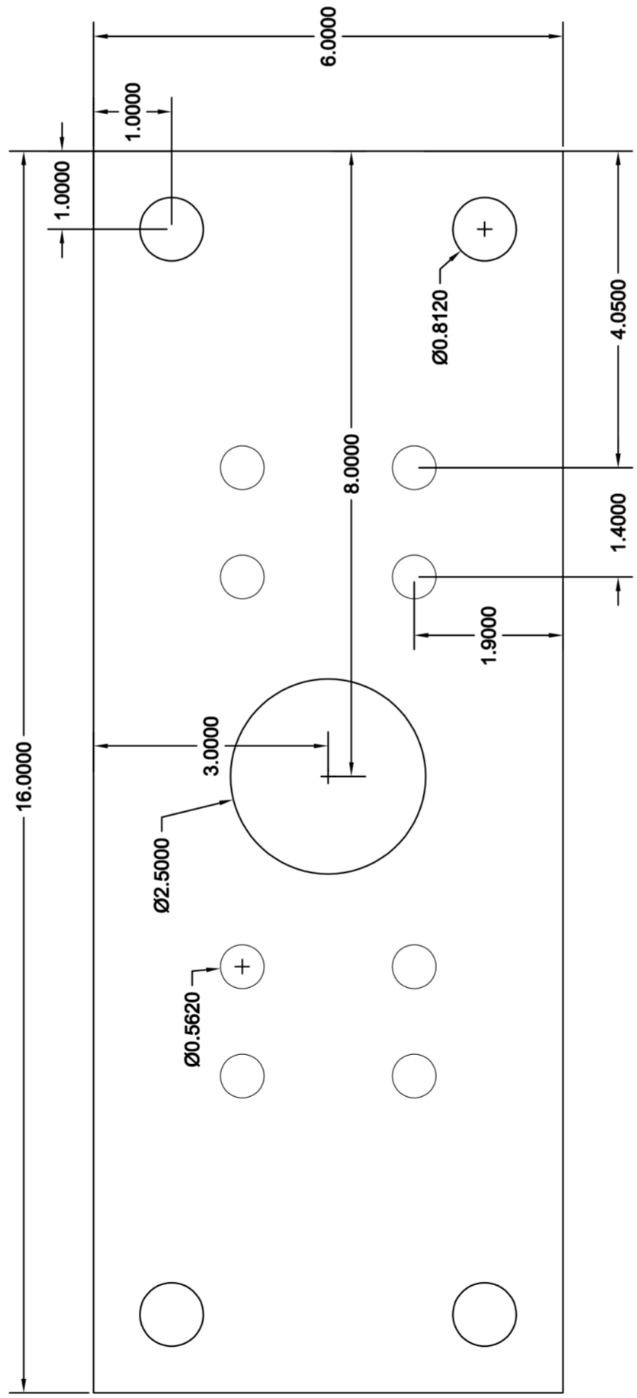


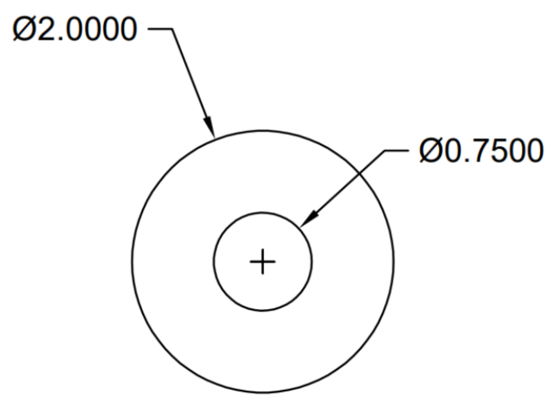
Figure C-9 Drawing of Part E.

Part F

All dimensions are in inches.
Quantity: 4 piece.
Material: Cold rolled steel.
Height= 18 in.
Diameter= 2 in.

3D Views

Top view



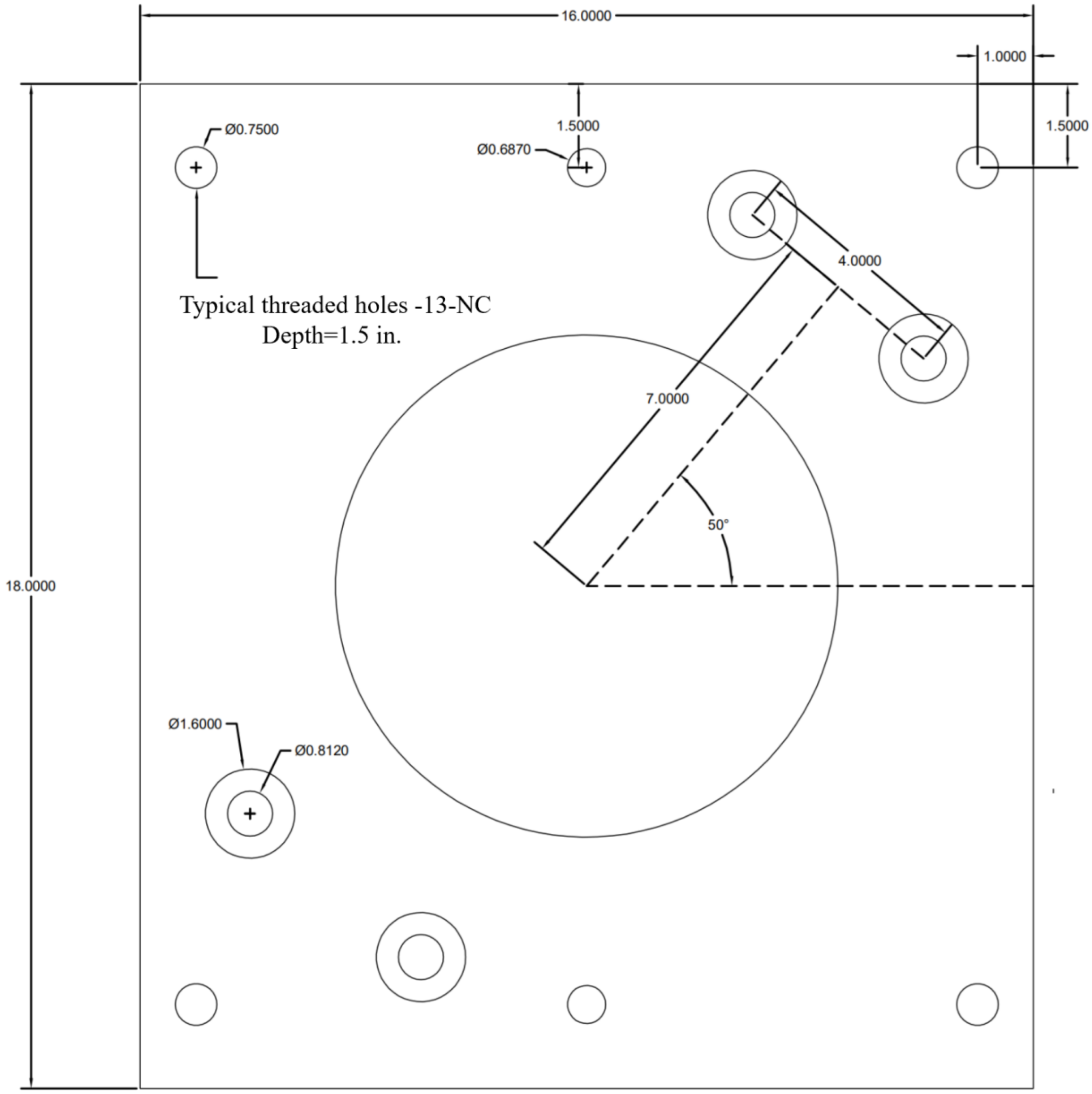
Typical threaded holes -10-NC-Depth=1.5 in.
Both sides of the column.

Figure C-10 Drawing of Part F.

Part G

All dimensions are in inches.
Quantity: 1 piece.
Material: Cold rolled steel.
Plate 18 in. × 16 in.
Thickness= 1.5 in.

Top view



Side view

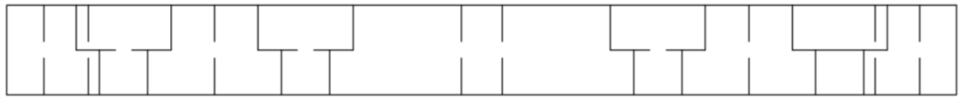


Figure C-11 Drawing of Part G.

Part G

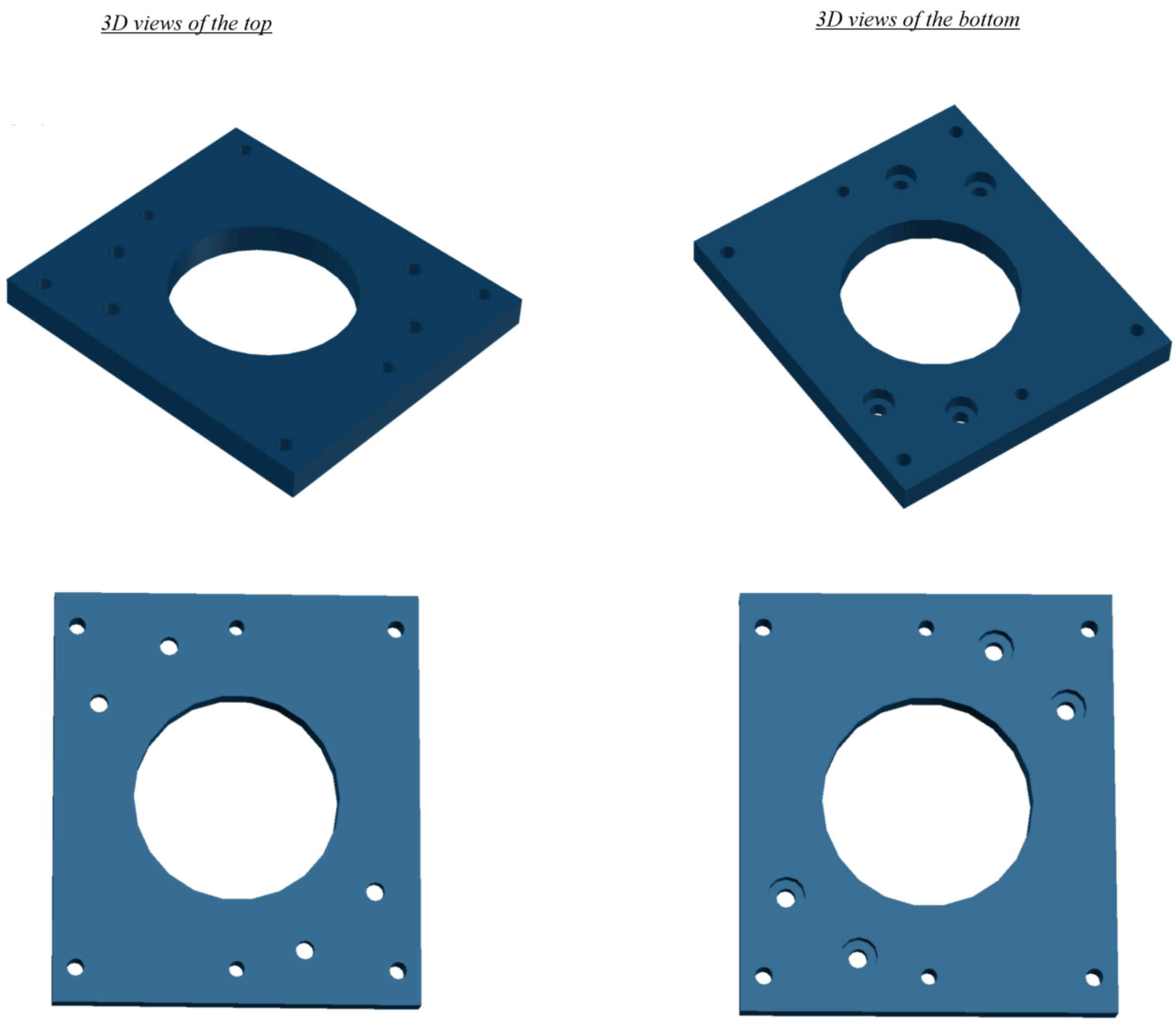


Figure C-12 3D views of Part G.

Appendix D

Unknowns of the first general form of the weight function, m_1

The stress intensity factor can be determined using the weight function method as follows:

$$K = \int_0^a \sigma(x) m(x,a) dx$$

where $\sigma(x)$ is the stress distribution and $m(x,a)$ is the weight function. The first evaluated form for the general equation of the weight function with three terms is as follows:

$$m_1(x,a) = \frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_2\left(1-\frac{x}{a}\right)\right]$$

In this equation, M_1 and M_2 should be determined. Two reference stress intensity factors for tension and bending loading are used to write two equations and determine the unknowns. The reference stress intensity factors and corresponding loadings are as follows:

$$K_{r1} = \sigma_0\sqrt{\pi a}\,\gamma_1 \quad for\ tension\ \sigma(x) = \sigma_0$$

$$K_{r2} = \sigma_0\sqrt{\pi a}\,\gamma_2 \quad for\ bending\ \sigma(x) = \sigma_0(1-\frac{x}{r})$$

Two equations can be written as follows:

$$\sigma_0\sqrt{\pi a}\gamma_1 = \int_0^a \sigma_0 \frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_2\left(1-\frac{x}{a}\right)\right]dx \qquad \textbf{(I)}$$

$$\sigma_0\sqrt{\pi a}\gamma_2 = \int_0^a \sigma_0(1-\frac{x}{r}) \frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_2\left(1-\frac{x}{a}\right)\right]dx \qquad \textbf{(II)}$$

Simplifying the first equation (I):

$$\sigma_0\sqrt{\pi a}\gamma_1 = \int_0^a \sigma_0 \frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_2\left(1-\frac{x}{a}\right)\right]dx$$

$$\sigma_0\sqrt{\pi a}\gamma_1 = \int_0^a \sigma_0 \frac{2}{\sqrt{2\pi a\left(1-\frac{x}{a}\right)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_2\left(1-\frac{x}{a}\right)\right]dx$$

$$\sigma_0\sqrt{\pi a}\gamma_1 = \int_0^a \sigma_0 \frac{2}{\sqrt{2\pi a}}\left[\left(1-\frac{x}{a}\right)^{-\frac{1}{2}} + M_1 \ + M_2\left(1-\frac{x}{a}\right)^{\frac{1}{2}}\right]dx$$

$$\sigma_0\sqrt{\pi a}\gamma_1 = \sigma_0 \frac{2}{\sqrt{2\pi a}}\left[\left(\frac{1}{1-\frac{1}{2}}\right)(-a)\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_1 x \ + \left(\frac{1}{1+\frac{1}{2}}\right)(-a)M_2\left(1-\frac{x}{a}\right)^{\frac{3}{2}}\right]|_0^a$$

$$\frac{\pi a\gamma_1}{\sqrt{2}} = \left[(2)(-a)\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_1 x \ + \left(\frac{2}{3}\right)(-a)M_2\left(1-\frac{x}{a}\right)^{\frac{3}{2}}\right]|_0^a$$

$$\frac{\pi a\gamma_1}{\sqrt{2}} = \left[(2)(-a)\left(1-\frac{a}{a}\right)^{\frac{1}{2}} + M_1(a) \ + \left(\frac{2}{3}\right)(-a)M_2\left(1-\frac{a}{a}\right)^{\frac{3}{2}}\right]$$

$$-\left[(2)(-a)\left(1-\frac{0}{a}\right)^{\frac{1}{2}} + M_1(0) \ + \left(\frac{2}{3}\right)(-a)M_2\left(1-\frac{0}{a}\right)^{\frac{3}{2}}\right]$$

$$\frac{\pi a\gamma_1}{\sqrt{2}} = [M_1(a)\ ] - \left[(2)(-a)(1)^{\frac{1}{2}} \ + \left(\frac{2}{3}\right)(-a)M_2(1)^{\frac{3}{2}}\right]$$

$$\frac{\pi a\gamma_1}{\sqrt{2}} = [M_1(a)\ ] - \left[(2)(-a) \ + \left(\frac{2}{3}\right)(-a)M_2\right]$$

$$\frac{\pi a\gamma_1}{\sqrt{2}} = [M_1(a\ )] - \left[(2)(-a) \ + \left(\frac{2}{3}\right)(-a)M_2\right]$$

$$\frac{\pi\gamma_1}{\sqrt{2}} = M_1 + \frac{2}{3}M_2 + 2 \quad \textbf{(III)}$$

Simplifying the second equation (II):

$$\sigma_0\sqrt{\pi a}\gamma_2 = \int_0^a \sigma_0(1-\frac{x}{r})\frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^{\frac{1}{2}} \ + M_2\left(1-\frac{x}{a}\right)\right]dx$$

$$\frac{\pi a\gamma_2}{\sqrt{2}} = \int_0^a (1-\frac{x}{r})\left[\left(1-\frac{x}{a}\right)^{-\frac{1}{2}} + M_1 + M_2\left(1-\frac{x}{a}\right)^{\frac{1}{2}}\right]dx$$

$$u = 1-\frac{x}{a} \qquad x = a - au \qquad dx = -adu$$

$$x = 0 \quad u = 1$$

$$x = a \quad u = 0$$

$$\frac{\pi a \gamma_2}{\sqrt{2}} = -\int_0^1 (1 - \frac{(a-au)}{r}) \left[u^{-\frac{1}{2}} + M_1 + M_2 u^{\frac{1}{2}}\right] (-a) du$$

$$\frac{\pi a \gamma_2}{\sqrt{2}} = (a) \int_0^1 \left(1 - \frac{a}{r} + \left(\frac{a}{r}\right) u\right) \left[u^{-\frac{1}{2}} + M_1 + M_2 u^{\frac{1}{2}}\right] du$$

$$(a) \int_0^1 \left(1 - \frac{a}{r} + \left(\frac{a}{r}\right) u\right) \left[u^{-\frac{1}{2}} + M_1 + M_2 u^{\frac{1}{2}}\right] du$$

$$= (a) \left(1 - \frac{a}{r}\right) \int_0^1 \left[u^{-\frac{1}{2}} + M_1 + M_2 u^{\frac{1}{2}}\right] du + (a) \left(\frac{a}{r}\right) \int_0^1 \left[u^{\frac{1}{2}} + M_1 u + M_2 u^{\frac{3}{2}}\right] du$$

$$= a \left(1 - \frac{a}{r}\right) \left[\frac{1}{1 - \frac{1}{2}} u^{\frac{1}{2}} + M_1 u + \frac{1}{1 + \frac{1}{2}} M_2 u^{\frac{3}{2}}\right] |_0^1$$

$$+ (a) \left(\frac{a}{r}\right) \left[\frac{1}{1 + \frac{1}{2}} u^{\frac{3}{2}} + \frac{1}{1+1} M_1 u^2 + \frac{1}{1 + \frac{3}{2}} M_2 u^{\frac{5}{2}}\right] |_0^1$$

$$= a \left(1 - \frac{a}{r}\right) \left[(2)(1) + M_1(1) + \left(\frac{2}{3}\right)(1) M_2\right] + (a) \left(\frac{a}{r}\right) \left[\left(\frac{2}{3}\right)(1) + \left(\frac{1}{2}\right) M_1(1) + \left(\frac{2}{5}\right) M_2\right]$$

$$= a \left(1 - \frac{a}{r}\right) \left[2 + M_1 + \frac{2}{3} M_2\right] + (a) \left(\frac{a}{r}\right) \left[\frac{2}{3} + \frac{1}{2} M_1 + \frac{2}{5} M_2\right]$$

$$= \left(a - \frac{a^2}{r}\right) \left[2 + M_1 + \frac{2}{3} M_2\right] + \left(\frac{a^2}{r}\right) \left[\frac{2}{3} + \frac{1}{2} M_1 + \frac{2}{5} M_2\right]$$

$$= \left(\frac{a^2}{2r} + a - \frac{a^2}{r}\right) M_1 + \left(\frac{2}{3} a - \frac{2a^2}{3r} + \frac{2a^2}{5r}\right) M_2 + \left(2a - \frac{2a^2}{r} + \frac{2a^2}{3r}\right)$$

$$\frac{\pi a \gamma_2}{\sqrt{2}} = \left(-\frac{a^2}{2r} + a\right) M_1 + \left(-\frac{4a^2}{15r} + \frac{2a}{3}\right) M_2 + \left(-\frac{4a^2}{3r} + 2a\right)$$

$$\frac{\pi \gamma_2}{\sqrt{2}} = \left(-\frac{a}{2r} + 1\right) M_1 + \left(-\frac{4a}{15r} + \frac{2}{3}\right) M_2 + \left(-\frac{4a}{3r} + 2\right) \qquad \textbf{(IV)}$$

The simplified equations (III and IV) can be now used to determine the unknowns. From equation (III), M_1 can be written as follows:

$$M_1 = \frac{\pi\gamma_1}{\sqrt{2}} - \frac{2}{3}M_2 - 2 \quad \textbf{(V)}$$

Using Equation (V) for M_1, Equation (IV) can be written as follows:

$$\frac{\pi\gamma_2}{\sqrt{2}} = \left(-\frac{a}{2r} + 1\right)\left(\frac{\pi\gamma_1}{\sqrt{2}} - \frac{2}{3}M_2 - 2\right) + \left(-\frac{4a}{15r} + \frac{2}{3}\right)M_2 + \left(-\frac{4a}{3r} + 2\right) \qquad \textbf{(VI)}$$

M_2 can be determined by simplifying Equation (VI):

$$\frac{\pi\gamma_2}{\sqrt{2}} = \left(-\frac{a}{2r} + 1\right)\left(\frac{\pi\gamma_1}{\sqrt{2}}\right) + \left(-\frac{a}{2r} + 1\right)\left(-\frac{2}{3}M_2\right) + \left(-\frac{a}{2r} + 1\right)(-2) + \left(-\frac{4a}{15r} + \frac{2}{3}\right)M_2 + \left(-\frac{4a}{3r} + 2\right)$$

$$\frac{\pi\gamma_2}{\sqrt{2}} = \left(-\frac{a}{2r} + 1\right)\left(\frac{\pi\gamma_1}{\sqrt{2}}\right) + \left(\frac{a}{3r} - \frac{2}{3} - \frac{4a}{15r} + \frac{2}{3}\right)M_2 + \left(-\frac{4a}{3r} + 2 + \frac{a}{r} - 2\right)$$

$$\frac{\pi\gamma_2}{\sqrt{2}} = \left(-\frac{a}{2r} + 1\right)\left(\frac{\pi\gamma_1}{\sqrt{2}}\right) + \left(\frac{a}{15r}\right)M_2 + \left(-\frac{a}{3r}\right)$$

$$\frac{\pi\gamma_2}{\sqrt{2}} - \left(-\frac{a}{2r} + 1\right)\left(\frac{\pi\gamma_1}{\sqrt{2}}\right) + \frac{a}{3r} = \left(\frac{a}{15r}\right)M_2$$

$$M_2 = \left(\frac{15r}{a}\right)\frac{\pi\gamma_2}{\sqrt{2}} + \left(\frac{15}{2} - \frac{15r}{a}\right)\left(\frac{\pi\gamma_1}{\sqrt{2}}\right) + 5$$

Therefore, the unknowns of the first form of the general weight function can be determined as follows:

$$M_2 = \left(\frac{15r}{a}\right)\frac{\pi\gamma_2}{\sqrt{2}} + \left(\frac{15}{2} - \frac{15r}{a}\right)\left(\frac{\pi\gamma_1}{\sqrt{2}}\right) + 5$$

$$M_1 = \frac{\pi\gamma_1}{\sqrt{2}} - \frac{2}{3}M_2 - 2$$

Unknowns of the second general form of the weight function, m_2

The stress intensity factor can be determined using the weight function method as follows:

$$K = \int_0^a \sigma(x) m(x,a) dx$$

where $\sigma(x)$ is the stress distribution and $m(x,a)$ is the weight function. The second evaluated form for the general equation of the weight function with three terms is as follows:

$$m_1(x,a) = \frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^1 + M_2\left(1-\frac{x}{a}\right)^2\right]$$

In this equation, M_1 and M_2 should be determined. Two reference stress intensity factors for tension and bending loading are used to write two equations and determine the unknowns. The reference stress intensity factors and corresponding loadings are as follows:

$$K_{r1} = \sigma_0\sqrt{\pi a}\,\gamma_1 \quad for\ tension\ \sigma(x) = \sigma_0$$

$$K_{r2} = \sigma_0\sqrt{\pi a}\,\gamma_2 \quad for\ bending\ \sigma(x) = \sigma_0(1-\frac{x}{r})$$

Two equations can be written as follows:

$$\sigma_0\sqrt{\pi a}\gamma_1 = \int_0^a \sigma_0 \frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^1 + M_2\left(1-\frac{x}{a}\right)^2\right]dx \qquad \textbf{(I)}$$

$$\sigma_0\sqrt{\pi a}\gamma_2 = \int_0^a \sigma_0(1-\frac{x}{r}) \frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^1 + M_2\left(1-\frac{x}{a}\right)^2\right]dx \qquad \textbf{(II)}$$

Simplifying the first equation (I):

$$\sigma_0\sqrt{\pi a}\gamma_1 = \int_0^a \sigma_0 \frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^1 + M_2\left(1-\frac{x}{a}\right)^2\right]dx$$

$$\sigma_0\sqrt{\pi a}\gamma_1 = \int_0^a \sigma_0 \frac{2}{\sqrt{2\pi a\left(1-\frac{x}{a}\right)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^1 + M_2\left(1-\frac{x}{a}\right)^2\right]dx$$

$$\sigma_0\sqrt{\pi a}\gamma_1 = \int_0^a \sigma_0 \frac{2}{\sqrt{2\pi a}}\left[\left(1-\frac{x}{a}\right)^{-\frac{1}{2}} + M_1\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_2\left(1-\frac{x}{a}\right)^{\frac{3}{2}}\right]dx$$

$$\frac{\pi a\gamma_1}{\sqrt{2}} = \left[\left(\frac{1}{1-\frac{1}{2}}\right)(-a)\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + M_1\left(\frac{1}{1+\frac{1}{2}}\right)(-a)\left(1-\frac{x}{a}\right)^{\frac{3}{2}} + \left(\frac{1}{1+\frac{3}{2}}\right)(-a)M_2\left(1-\frac{x}{a}\right)^{\frac{5}{2}}\right]\Big|_0^a$$

$$\frac{\pi a\gamma_1}{\sqrt{2}} = \left[(2)(-a)\left(1-\frac{x}{a}\right)^{\frac{1}{2}} + \left(\frac{2}{3}\right)(-a)M_1\left(1-\frac{x}{a}\right)^{\frac{3}{2}} + \left(\frac{2}{5}\right)(-a)M_2\left(1-\frac{x}{a}\right)^{\frac{5}{2}}\right]\Big|_0^a$$

$$\frac{\pi a\gamma_1}{\sqrt{2}} = \left[2a + \frac{2}{3}M_1 a + \frac{2}{5}M_2 a\right]$$

$$\frac{\pi\gamma_1}{\sqrt{2}} = \left[2 + \frac{2}{3}M_1 + \frac{2}{5}M_2\right] \qquad \textbf{(III)}$$

Simplifying the second equation (II):

$$\sigma_0\sqrt{\pi a}\gamma_2 = \int_0^a \sigma_0\left(1-\frac{x}{r}\right)\frac{2}{\sqrt{2\pi(a-x)}}\left[1 + M_1\left(1-\frac{x}{a}\right)^1 + M_2\left(1-\frac{x}{a}\right)^2\right]dx$$

$$\frac{\pi a\gamma_2}{\sqrt{2}} = \int_0^a \left(1-\frac{x}{r}\right)\left[1 + M_1\left(1-\frac{x}{a}\right)^1 + M_2\left(1-\frac{x}{a}\right)^2\right]dx$$

$$u = 1-\frac{x}{a} \qquad x = a - au \qquad dx = -adu$$

$$x = 0 \quad u = 1$$

$$x = a \quad u = 0$$

$$\frac{\pi a\gamma_2}{\sqrt{2}} = -\int_0^1 \left(1-\frac{(a-au)}{r}\right)\left[u^{-\frac{1}{2}} + M_1 u^{\frac{1}{2}} + M_2 u^{\frac{3}{2}}\right](-a)du$$

$$\frac{\pi\gamma_2}{\sqrt{2}} = \int_0^1 \left(1-\frac{a}{r}+\left(\frac{a}{r}\right)u\right)\left[u^{-\frac{1}{2}} + M_1 u^{\frac{1}{2}} + M_2 u^{\frac{3}{2}}\right]du$$

$$\int_0^1 \left(1-\frac{a}{r}+\left(\frac{a}{r}\right)u\right)\left[u^{-\frac{1}{2}} + M_1 u^{\frac{1}{2}} + M_2 u^{\frac{3}{2}}\right]du$$

$$= \left(1-\frac{a}{r}\right)\int_0^1 \left[u^{-\frac{1}{2}} + M_1 u^{\frac{1}{2}} + M_2 u^{\frac{3}{2}}\right]du + \left(\frac{a}{r}\right)\int_0^1 \left[u^{\frac{1}{2}} + M_1 u^{\frac{3}{2}} + M_2 u^{\frac{5}{2}}\right]du$$

$$= \left(1-\frac{a}{r}\right)\left[\frac{1}{1-\frac{1}{2}}u^{\frac{1}{2}} + \frac{1}{1+\frac{1}{2}}M_1u^{\frac{3}{2}} + \frac{1}{1+\frac{3}{2}}M_2u^{\frac{5}{2}}\right]\Big|_0^1$$

$$+ \left(\frac{a}{r}\right)\left[\frac{1}{1+\frac{1}{2}}u^{\frac{3}{2}} + \frac{1}{1+\frac{3}{2}}M_1u^{\frac{5}{2}} + \frac{1}{1+\frac{5}{2}}M_2u^{\frac{7}{2}}\right]\Big|_0^1$$

$$= \left(1-\frac{a}{r}\right)\left[2u^{\frac{1}{2}} + \frac{2}{3}M_1u^{\frac{3}{2}} + \frac{2}{5}M_2u^{\frac{5}{2}}\right]\Big|_0^1 + \left(\frac{a}{r}\right)\left[\frac{2}{3}u^{\frac{3}{2}} + \frac{2}{5}M_1u^{\frac{5}{2}} + \frac{2}{7}M_2u^{\frac{7}{2}}\right]\Big|_0^1$$

$$= \left(1-\frac{a}{r}\right)\left[(2)(1) + \frac{2}{3}M_1(1) + \left(\frac{2}{5}\right)(1)M_2\right] + \left(\frac{a}{r}\right)\left[\left(\frac{2}{3}\right)(1) + \left(\frac{2}{5}\right)M_1(1) + \left(\frac{2}{7}\right)M_2\right]$$

$$= \left(1-\frac{a}{r}\right)\left[2 + \frac{2}{3}M_1 + \frac{2}{5}M_2\right] + \left(\frac{a}{r}\right)\left[\frac{2}{3} + \frac{2}{5}M_1 + \frac{2}{7}M_2\right]$$

$$= \left(\frac{2}{3} - \frac{2a}{3r} + \frac{2a}{5r}\right)M_1 + \left(\frac{2}{5} - \frac{2a}{5r} + \frac{2a}{7r}\right)M_2 + \left(2 - \frac{2a}{r} + \frac{2a}{3r}\right)$$

$$= \left(\frac{2}{3} - \frac{4a}{15r}\right)M_1 + \left(\frac{2}{5} - \frac{4a}{35r}\right)M_2 + \left(2 - \frac{4a}{3r}\right)$$

$$\frac{\pi\gamma_2}{\sqrt{2}} = \left(\frac{2}{3} - \frac{4a}{15r}\right)M_1 + \left(\frac{2}{5} - \frac{4a}{35r}\right)M_2 + \left(2 - \frac{4a}{3r}\right)$$

$$\frac{\pi\gamma_2}{\sqrt{2}} = \left(\frac{2}{3} - \frac{4a}{15r}\right)M_1 + \left(\frac{2}{5} - \frac{4a}{35r}\right)M_2 + \left(2 - \frac{4a}{3r}\right) \qquad \textbf{(IV)}$$

The simplified equations (III and IV) can now be used to determine the unknowns. From equation (III), M_1can be written as follows:

$$M_1 = \left(\frac{3}{2}\right)\left(\frac{\pi\gamma_1}{\sqrt{2}} - \frac{2}{5}M_2 - 2\right) \quad \textbf{(V)}$$

Using this Equation (V) for M_1, Equation (IV) can be written as follows:

$$\frac{\pi\gamma_2}{\sqrt{2}} = \left(\frac{2}{3} - \frac{4a}{15r}\right)\left(\frac{3}{2}\right)\left(\frac{\pi\gamma_1}{\sqrt{2}} - \frac{2}{5}M_2 - 2\right) + \left(\frac{2}{5} - \frac{4a}{35r}\right)M_2 + \left(2 - \frac{4a}{3r}\right) \quad \textbf{(VI)}$$

M_2 can be determined by simplifying Equation (VI):

$$\frac{\pi\gamma_2}{\sqrt{2}} = \left(\frac{2}{3} - \frac{4a}{15r}\right)\left(\frac{3}{2}\right)\left(\frac{\pi\gamma_1}{\sqrt{2}} - \frac{2}{5}M_2 - 2\right) + \left(\frac{2}{5} - \frac{4a}{35r}\right)M_2 + \left(2 - \frac{4a}{3r}\right)$$

$$\frac{\pi\gamma_2}{\sqrt{2}} = \left(1 - \frac{2a}{5r}\right)\left(\frac{\pi\gamma_1}{\sqrt{2}}\right) + \left(1 - \frac{2a}{5r}\right)\left(-\frac{2}{5}M_2\right) + \left(1 - \frac{2a}{5r}\right)(-2) + \left(\frac{2}{5} - \frac{4a}{35r}\right)M_2 + \left(2 - \frac{4a}{3r}\right)$$

$$\frac{\pi\gamma_2}{\sqrt{2}} = \left(1 - \frac{2a}{5r}\right)\left(\frac{\pi\gamma_1}{\sqrt{2}}\right) + \left(\frac{8a}{175r}\right)M_2 + \left(-\frac{8a}{15r}\right)$$

$$M_2 = \frac{175r\pi\gamma_2}{8a\sqrt{2}} - \left(\frac{175r}{8a} - \frac{35}{4}\right)\left(\frac{\pi\gamma_1}{\sqrt{2}}\right) + \left(\frac{35}{3}\right)$$

Therefore, the unknowns of the second form of the general weight function can be determined as follows:

$$M_2 = \frac{175r\pi\gamma_2}{8a\sqrt{2}} - \left(\frac{175r}{8a} - \frac{35}{4}\right)\left(\frac{\pi\gamma_1}{\sqrt{2}}\right) + \left(\frac{35}{3}\right)$$

$$M_1 = \left(\frac{3}{2}\right)\left(\frac{\pi\gamma_1}{\sqrt{2}} - \frac{2}{5}M_2 - 2\right)$$

Appendix E

In this appendix, MDRM tables and graphs related to analysis of tests at the University of Waterloo are listed. The MDRM analysis of tests results is presented in Chapter 6.

Table E-1 Input gird for MDRM analysis of Test #1.

Variable	COF (X1)	σ'_f (X2)	ε'_f (X3)	SWT	N_f	Log(N_f)
1	0.3688	2183	1.99	2.310	31401331	7.497
2	0.4423	2183	1.99	2.946	4964485	6.696
3	0.5500	2183	1.99	3.924	578252	5.762
4	0.6577	2183	1.99	4.538	201159	5.304
5	0.7312	2183	1.99	4.890	118926	5.075
6	0.5500	1890	1.99	3.924	76772	4.885
7	0.5500	2037	1.99	3.924	213154	5.329
8	0.5500	2180	1.99	3.924	567724	5.754
9	0.5500	2333	1.99	3.924	1554601	6.192
10	0.5500	2515	1.99	3.924	4815299	6.683
11	0.5500	2183	1.2477	3.924	569473	5.755
12	0.5500	2183	1.5840	3.924	573461	5.759
13	0.5500	2183	1.9650	3.924	577958	5.762
14	0.5500	2183	2.4376	3.924	583508	5.766
15	0.5500	2183	3.0948	3.924	591174	5.772

Table E-2 MDRM analysis results for Test #1.

Moments	Entropy	i	0	1	2	3	4
m=1	2.1440	λ_i	2.059984	1.7508E-11			
		α_i		11.7759			
m=2	1.2278	λ_i	291.8176	-257.5104	7.5369		
		α_i		0.1597	1.0754		
m=3	1.1909	λ_i	7.06E+02	281.6782	275.1126	-765.2224	
		α_i		-2.1467	-2.2652	-0.0368	
m=4	1.2041	λ_i	56.10768	54.5335	2214.4578	-389.7796	-161.6226
		α_i		-0.2316	-3.2993	-3.2815	-0.2893

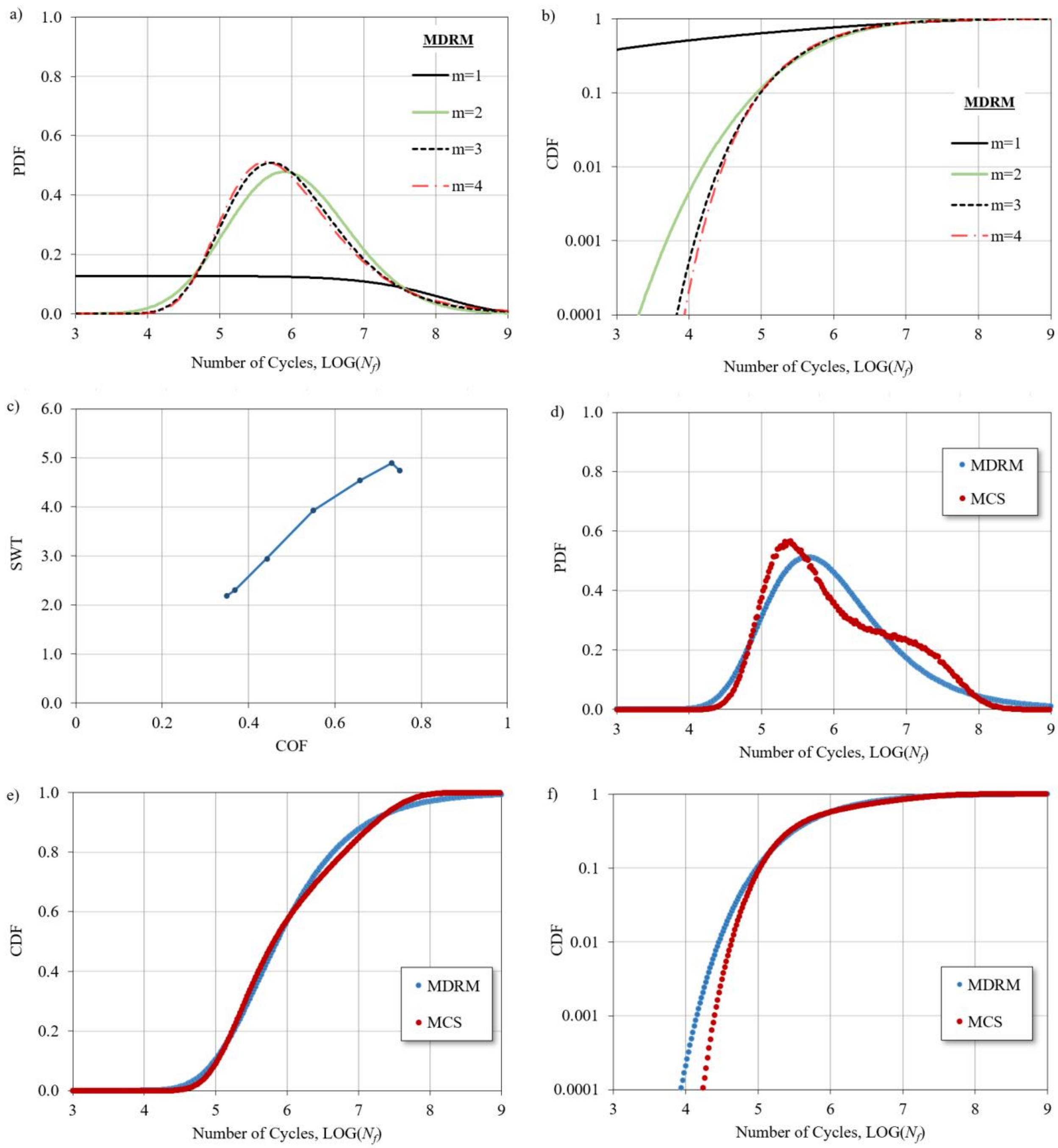


Figure E-1 Probabilistic analysis for Test #1: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale(f).

Table E-3 Input gird for MDRM analysis of Tests 2-3.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.5117	2183	1.99	3.883	624797	5.796
2	0.5577	2183	1.99	4.107	413672	5.617
3	0.6250	2183	1.99	4.551	196981	5.294
4	0.6923	2183	1.99	4.701	156671	5.195
5	0.7383	2183	1.99	4.742	147436	5.169
6	0.6250	1890	1.99	4.551	30525	4.485
7	0.6250	2037	1.99	4.551	76676	4.885
8	0.6250	2180	1.99	4.551	193532	5.287
9	0.6250	2333	1.99	4.551	515359	5.712
10	0.6250	2515	1.99	4.551	1573923	6.197
11	0.6250	2183	1.2477	4.551	190568	5.280
12	0.6250	2183	1.5840	4.551	193489	5.287
13	0.6250	2183	1.9650	4.551	196767	5.294
14	0.6250	2183	2.4376	4.551	200792	5.303
15	0.6250	2183	3.0948	4.551	206315	5.315

Table E-4 MDRM analysis results for Tests #2-3.

Moments	Entropy	i	0	1	2	3	4
m=1	1.9431	λ_i	1.87619	4.0662E-12			
		α_i		13.7028			
m=2	0.4436	λ_i	328.448	-228.2888	12.4192		
		α_i		0.4257	1.4344		
m=3	0.4484	λ_i	4.21E+01	88.9927	-133.6726	325.8238	
		α_i		1.0916	0.9161	-1.5558	
m=4	0.4334	λ_i	681.4341	-284.2440	-527.1800	277.3256	20.0713
		α_i		0.7169	0.2788	0.8247	-9.7058

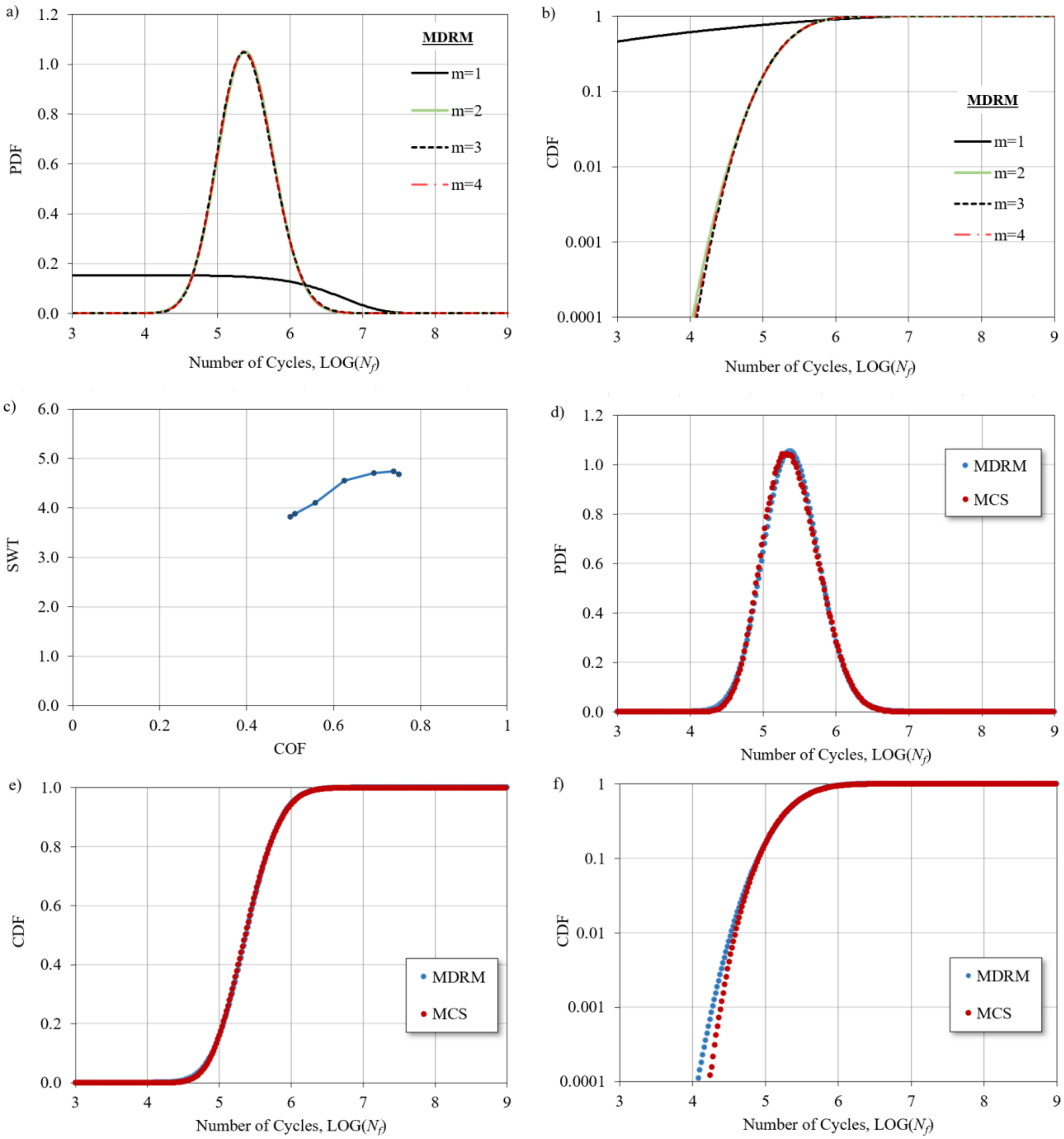


Figure E-2 Probabilistic analysis for Tests #2-3: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale(f).

Table E-5 Input gird for MDRM analysis of Tests 4-6.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.6738	2183	1.99	4.805	134403	5.128
2	0.6885	2183	1.99	4.753	145077	5.162
3	0.7100	2183	1.99	4.729	150205	5.177
4	0.7315	2183	1.99	4.840	127817	5.107
5	0.7462	2183	1.99	4.875	121457	5.084
6	0.7100	1890	1.99	4.729	24508	4.389
7	0.7100	2037	1.99	4.729	59634	4.775
8	0.7100	2180	1.99	4.729	147615	5.169
9	0.7100	2333	1.99	4.729	388801	5.590
10	0.7100	2515	1.99	4.729	1180545	6.072
11	0.7100	2183	1.2477	4.729	144312	5.159
12	0.7100	2183	1.5840	4.729	146999	5.167
13	0.7100	2183	1.9650	4.729	150009	5.176
14	0.7100	2183	2.4376	4.729	153698	5.187
15	0.7100	2183	3.0948	4.729	158749	5.201

Table E-6 MDRM analysis results for Tests #4-6.

Moments	Entropy	i	0	1	2	3	4
m=1	1.8774	λ_i	1.818851	1.9144E-12			
		α_i		14.5472			
m=2	0.2018	λ_i	388.2069	-254.6107	14.6905		
		α_i		0.4947	1.5433		
m=3	0.2084	λ_i	1.20E+02	-157.5919	270.2571	1.0512	
		α_i		0.2352	-0.8811	2.3312	
m=4	0.1959	λ_i	610.5378	-159.3546	-363.5875	264.9740	-166.4605
		α_i		0.5346	0.4536	1.0884	1.1183

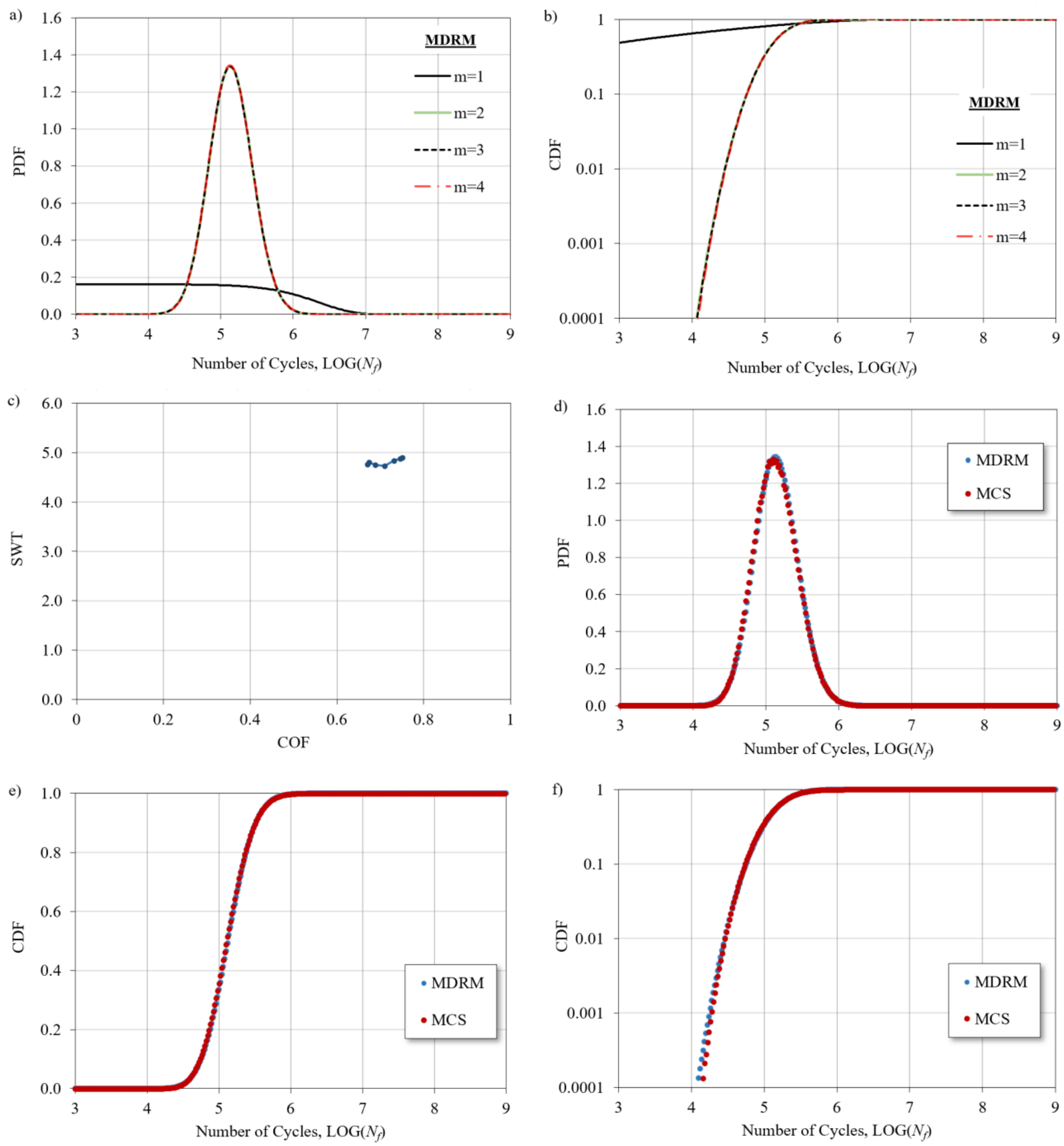


Figure E-3 Probabilistic analysis for Tests #4-6: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale(f).

Table E-7 Input gird for MDRM analysis of Tests 7-9.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.3592	2183	1.99	3.324	1994574	6.300
2	0.4346	2183	1.99	3.997	505279	5.704
3	0.5450	2183	1.99	4.672	163512	5.214
4	0.6554	2183	1.99	4.893	118433	5.073
5	0.7308	2183	1.99	4.799	135607	5.132
6	0.5450	1890	1.99	4.672	26237	4.419
7	0.5450	2037	1.99	4.672	64498	4.810
8	0.5450	2180	1.99	4.672	160679	5.206
9	0.5450	2333	1.99	4.672	424755	5.628
10	0.5450	2515	1.99	4.672	1292231	6.111
11	0.5450	2183	1.2477	4.672	157459	5.197
12	0.5450	2183	1.5840	4.672	160218	5.205
13	0.5450	2183	1.9650	4.672	163310	5.213
14	0.5450	2183	2.4376	4.672	167103	5.223
15	0.5450	2183	3.0948	4.672	172299	5.236

Table E-8 MDRM analysis results for Tests #7-9.

Moments	Entropy	i	0	1	2	3	4
m=1	1.9805	λ_i	1.894416	6.5868E-12			
		α_i		13.3172			
m=2	0.7242	λ_i	240.1839	-193.3622	27.2879		
		α_i		0.4426	1.0775		
m=3	0.7269	λ_i	1.74E+02	23.9058	24.0061	-133.3021	
		α_i		-2.8276	1.1449	0.5548	
m=4	0.7101	λ_i	497.6401	-243.0761	-407.1558	225.7778	74.8736
		α_i		0.4329	0.2657	0.6180	-1.7160

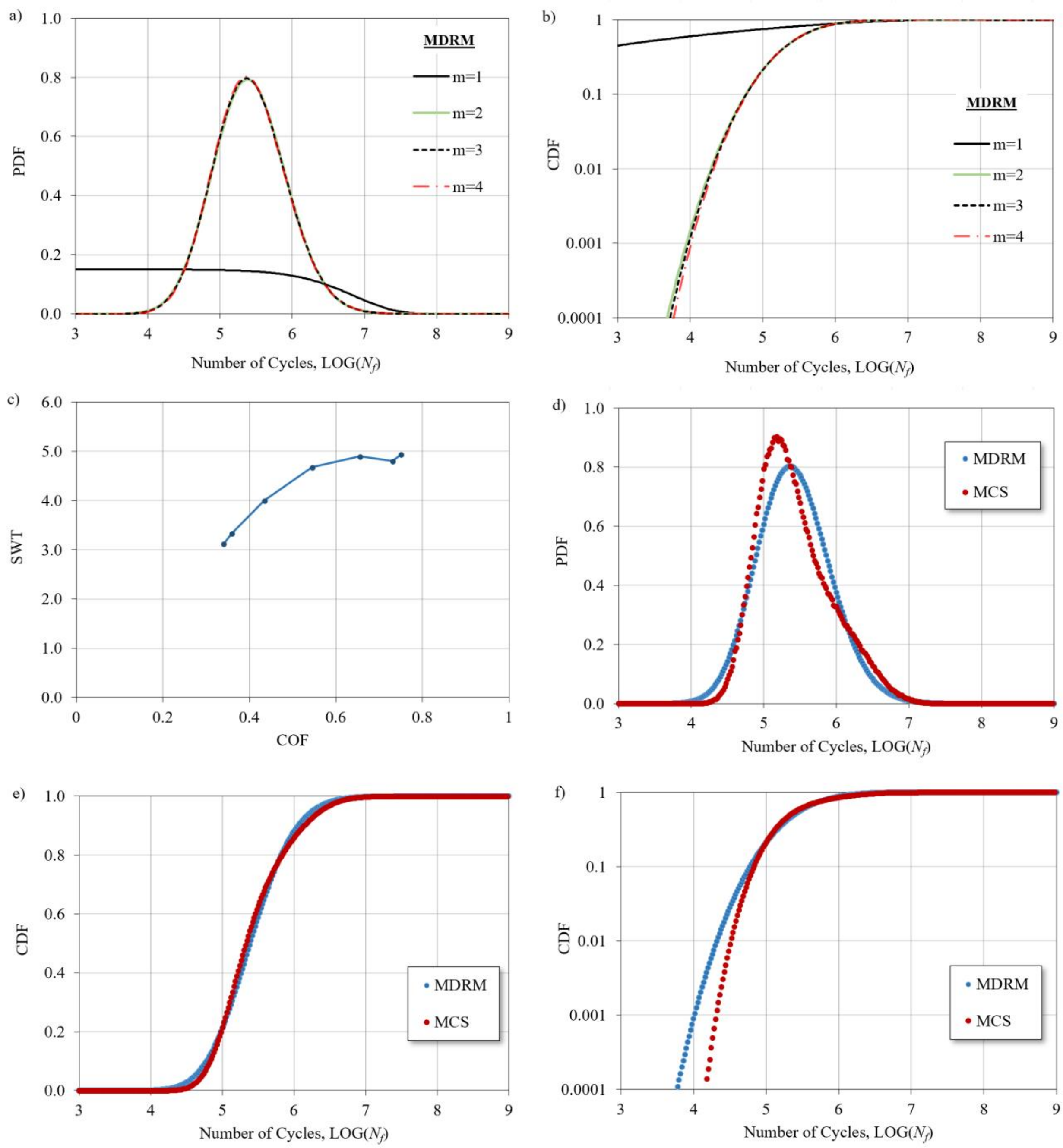


Figure E-4 Probabilistic analysis for Tests #7-9: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale (f).

Table E-9 Input gird for MDRM analysis of Tests 10-12.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.2544	2183	1.99	2.710	9333366	6.970
2	0.3500	2183	1.99	3.440	1541323	6.188
3	0.4900	2183	1.99	4.675	162811	5.212
4	0.6300	2183	1.99	5.124	86401	4.937
5	0.7256	2183	1.99	5.335	65959	4.819
6	0.4900	1890	1.99	4.675	26146	4.417
7	0.4900	2037	1.99	4.675	64242	4.808
8	0.4900	2180	1.99	4.675	159990	5.204
9	0.4900	2333	1.99	4.675	422859	5.626
10	0.4900	2515	1.99	4.675	1286337	6.109
11	0.4900	2183	1.2477	4.675	156765	5.195
12	0.4900	2183	1.5840	4.675	159521	5.203
13	0.4900	2183	1.9650	4.675	162609	5.211
14	0.4900	2183	2.4376	4.675	166396	5.221
15	0.4900	2183	3.0948	4.675	171585	5.234

Table E-10 MDRM analysis results for Tests #10-12.

Moments	Entropy	i	0	1	2	3	4
m=1	2.0708	λ_i	1.987338	4.6706E-11			
		α_i		11.7128			
m=2	1.1480	λ_i	339.4438	-306.8349	5.9744		
		α_i		0.1277	1.1531		
m=3	1.1015	λ_i	7.06E+02	83.7336	416.3637	-836.8818	
		α_i		0.1754	-2.1725	-0.0056	
m=4	1.1169	λ_i	45.42932	30.1902	558.5988	-39.9269	-104.7215
		α_i		0.3781	-2.5649	-2.7349	0.0233

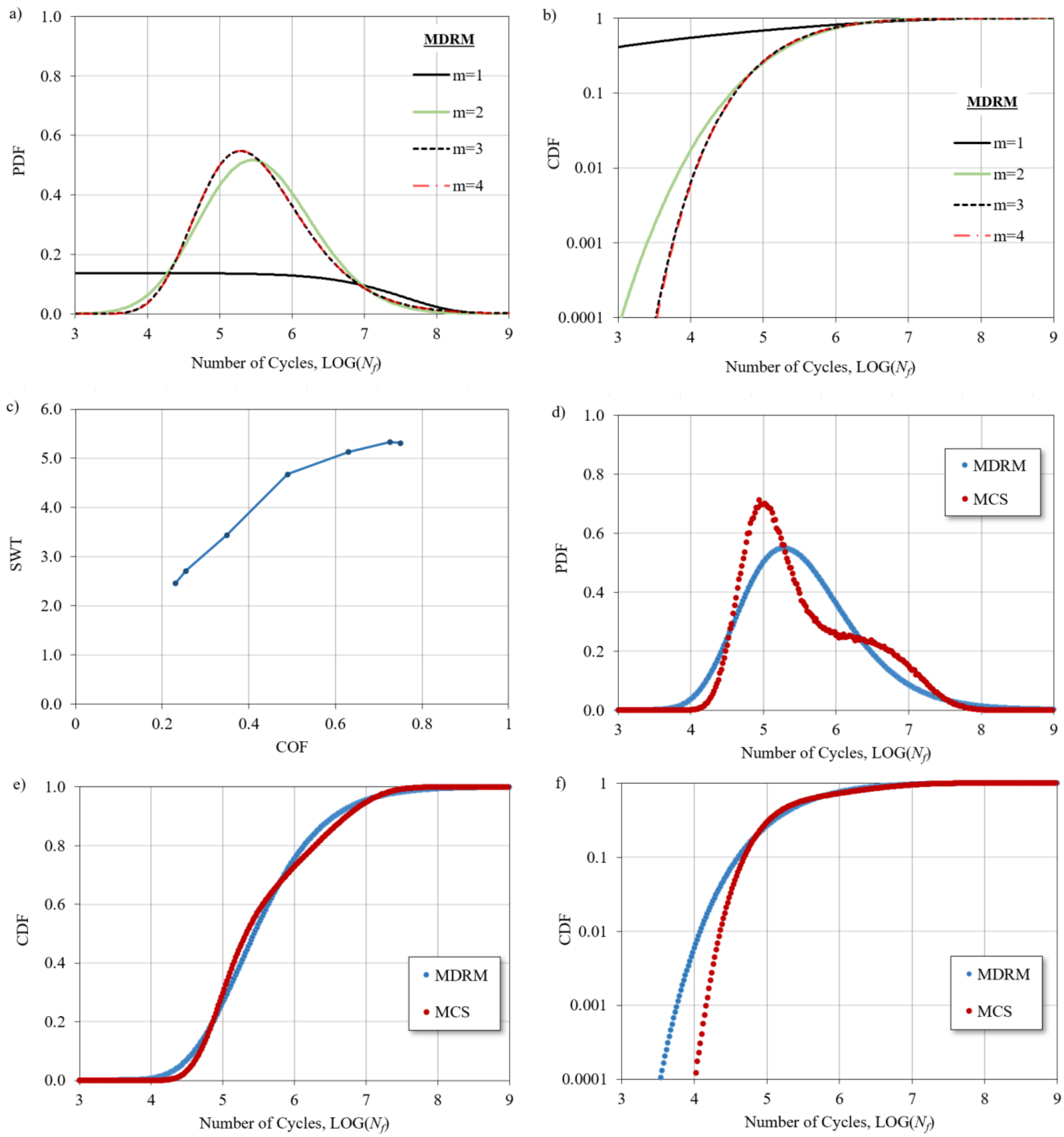


Figure E-5 Probabilistic analysis for Tests #10-12: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale (f).

Table E-11 Input gird for MDRM analysis of Tests #13-15.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.5308	2183	1.99	5.116	87286	4.941
2	0.5731	2183	1.99	5.348	64929	4.812
3	0.6350	2183	1.99	5.576	49472	4.694
4	0.6969	2183	1.99	5.764	40136	4.604
5	0.7392	2183	1.99	5.610	47601	4.678
6	0.6350	1890	1.99	5.576	10508	4.022
7	0.6350	2037	1.99	5.576	21974	4.342
8	0.6350	2180	1.99	5.576	48701	4.688
9	0.6350	2333	1.99	5.576	119398	5.077
10	0.6350	2515	1.99	5.576	347723	5.541
11	0.6350	2183	1.2477	5.576	45466	4.658
12	0.6350	2183	1.5840	5.576	47302	4.675
13	0.6350	2183	1.9650	5.576	49340	4.693
14	0.6350	2183	2.4376	5.576	51813	4.714
15	0.6350	2183	3.0948	5.576	55165	4.742

Table E-12 MDRM analysis results for Tests #13-15.

Moments	Entropy	i	0	1	2	3	4
m=1	1.7915	λ_i	1.714323	1.7393E-12			
		α_i		15.4882			
m=2	0.1806	λ_i	362.4911	-252.4419	15.2188		
		α_i		0.4720	1.5255		
m=3	0.1863	λ_i	1.98E+02	65.0113	2.1031	-112.3781	
		α_i		-2.8969	2.2322	0.5567	
m=4	0.1725	λ_i	705.505	115.9069	529.9024	213.5912	#########
		α_i		0.6097	-0.5858	-1.1041	0.0108

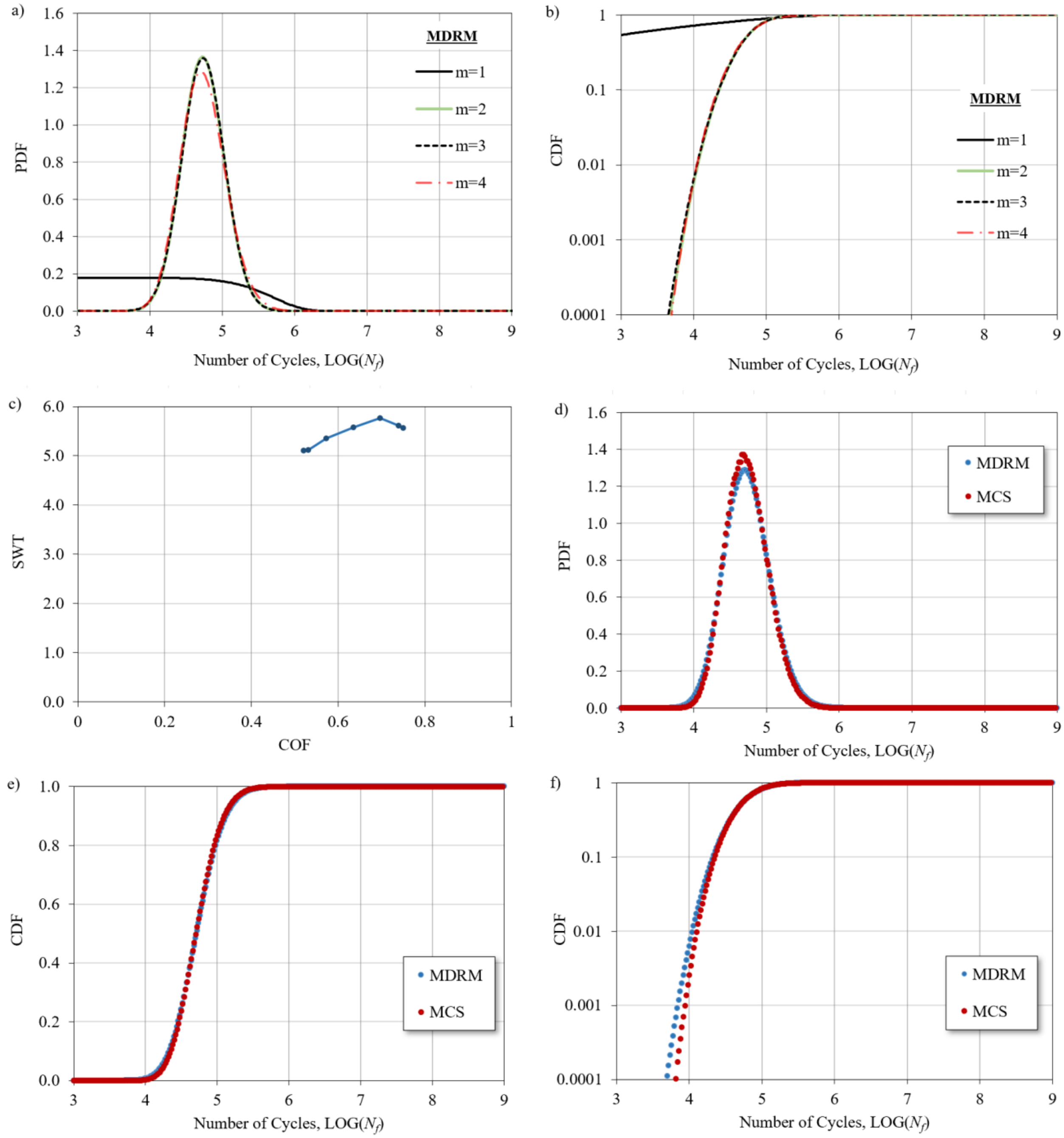


Figure E-6 Probabilistic analysis for Tests #13-15: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale (f).

Table E-13 Input gird for MDRM analysis of Tests #16-18.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.3688	2183	1.99	4.034	471641	5.674
2	0.4423	2183	1.99	4.774	140666	5.148
3	0.5500	2183	1.99	5.555	50671	4.705
4	0.6577	2183	1.99	5.699	43087	4.634
5	0.7312	2183	1.99	5.868	35903	4.555
6	0.5500	1890	1.99	5.555	10693	4.029
7	0.5500	2037	1.99	5.555	22439	4.351
8	0.5500	2180	1.99	5.555	49878	4.698
9	0.5500	2333	1.99	5.555	122544	5.088
10	0.5500	2515	1.99	5.555	357358	5.553
11	0.5500	2183	1.2477	5.555	46627	4.669
12	0.5500	2183	1.5840	5.555	48480	4.686
13	0.5500	2183	1.9650	5.555	50537	4.704
14	0.5500	2183	2.4376	5.555	53034	4.725
15	0.5500	2183	3.0948	5.555	56418	4.751

Table E-14 MDRM analysis results for Tests #16-18.

Moments	Entropy	i	0	1	2	3	4
m=1	1.8749	λ_i	1.792692	9.6606E-12			
		α_i		13.8580			
m=2	0.6128	λ_i	231.8744	-194.3566	31.2920		
		α_i		0.4654	1.0848		
m=3	0.6087	λ_i	3.12E+01	194.6314	-97.8701	63.7359	
		α_i		-1.7042	0.8213	1.0094	
m=4	0.6006	λ_i	526.688	49.2274	35.7343	-535.4082	23.9883
		α_i		0.8354	0.6762	0.2685	-1.2209

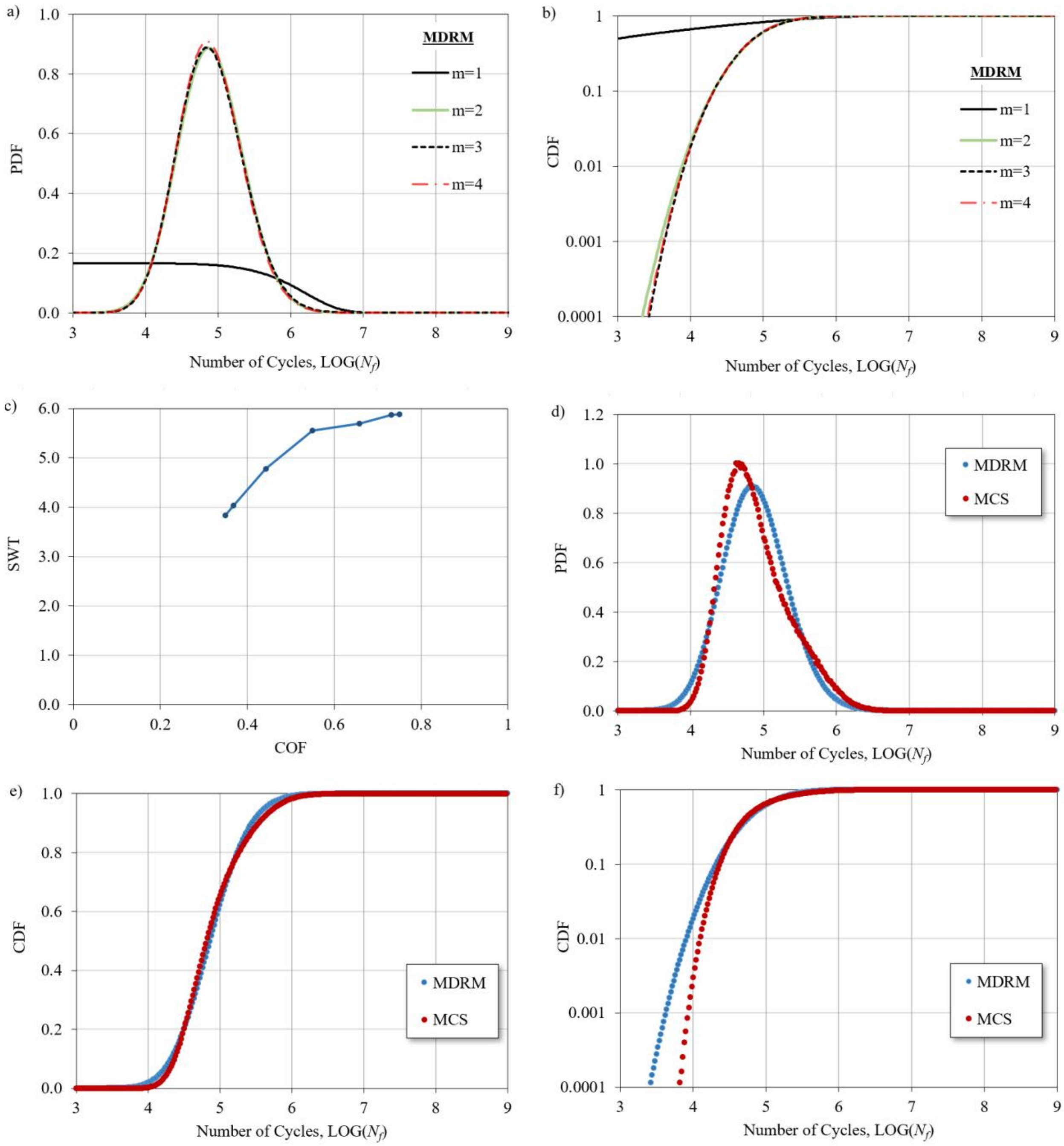


Figure E-7 Probabilistic analysis for Tests #16-18: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale (f).

Table E-15 Input gird for MDRM analysis of Tests #19-21.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.2830	2183	1.99	3.433	1565548	6.195
2	0.3731	2183	1.99	4.434	237531	5.376
3	0.5050	2183	1.99	5.648	45610	4.659
4	0.6369	2183	1.99	5.980	31999	4.505
5	0.7270	2183	1.99	6.066	29344	4.468
6	0.5050	1890	1.99	5.648	1565548	6.195
7	0.5050	2037	1.99	5.648	237531	5.376
8	0.5050	2180	1.99	5.648	45610	4.659
9	0.5050	2333	1.99	5.648	31999	4.505
10	0.5050	2515	1.99	5.648	29344	4.468
11	0.5050	2183	1.2477	5.648	41728	4.620
12	0.5050	2183	1.5840	5.648	43508	4.639
13	0.5050	2183	1.9650	5.648	45482	4.658
14	0.5050	2183	2.4376	5.648	47875	4.680
15	0.5050	2183	3.0948	5.648	51117	4.709

Table E-16 MDRM analysis results for Tests #19-21.

Moments	Entropy	i	0	1	2	3	4
m=1	1.9453	λ_i	1.869174	4.6094E-11			
		α_i		12.4583			
m=2	0.9362	λ_i	187.5239	-169.3400	23.3232		
		α_i		0.3462	0.9555		
m=3	0.8794	λ_i	7.06E+02	382.9117	-862.9374	74.3352	
		α_i		-1.8740	-0.0492	-0.0148	
m=4	0.8853	λ_i	76.49429	994.1446	-499.7818	424.0000	-411.6519
		α_i		-2.1243	-0.4905	-0.6098	-1.4397

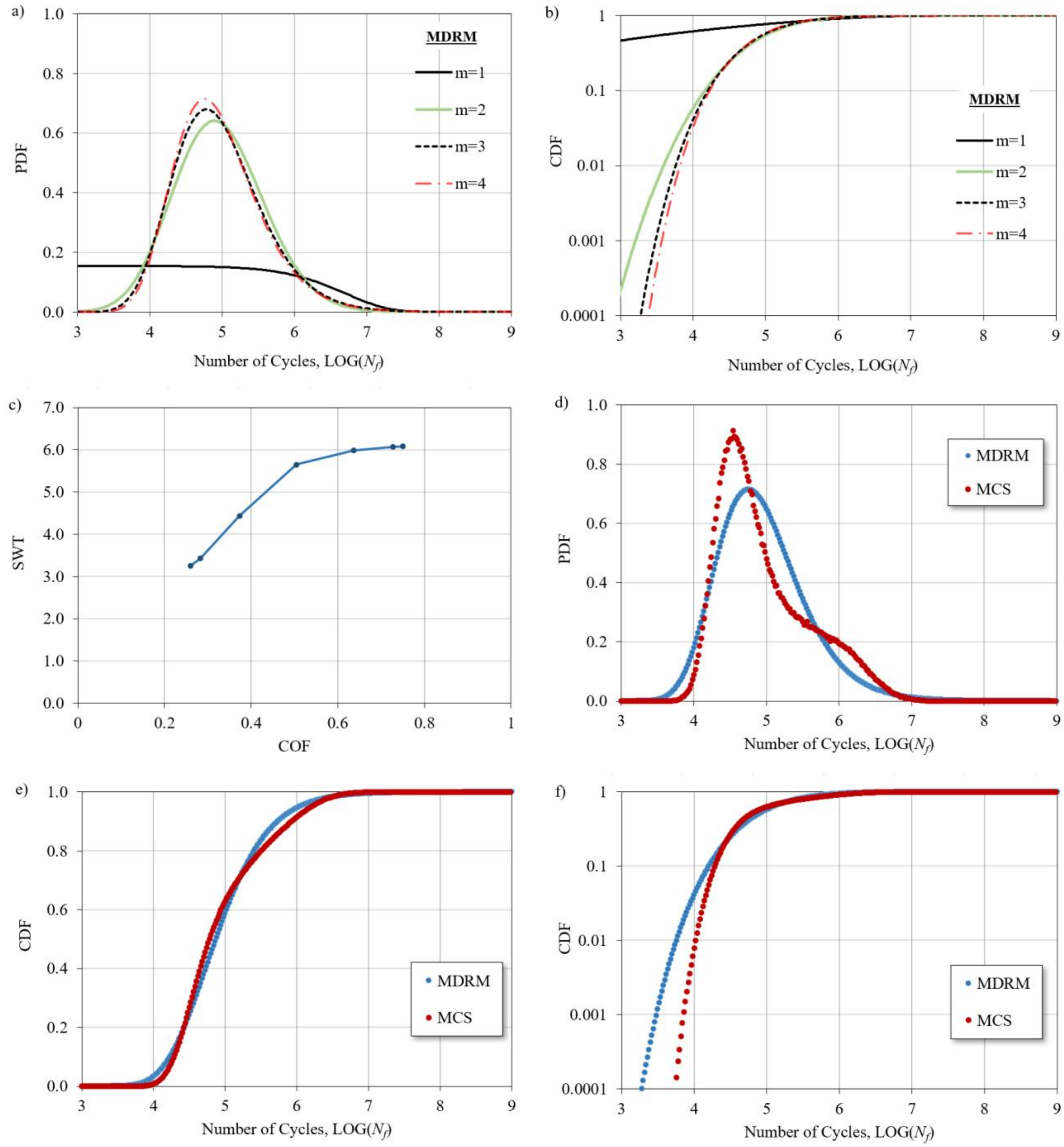


Figure E-8 Probabilistic analysis for Tests #19-21: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale (f).

Table E-17 Input gird for MDRM analysis of Tests #22-23.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.6356	2675	0.2067	4.369	135959	5.133
2	0.6577	2675	0.2067	4.311	145091	5.162
3	0.6900	2675	0.2067	4.472	121325	5.084
4	0.7223	2675	0.2067	4.560	110407	5.043
5	0.7444	2675	0.2067	4.480	120355	5.080
6	0.6900	2316.23	0.2067	4.472	34964	4.544
7	0.6900	2496.68	0.2067	4.472	65207	4.814
8	0.6900	2671.66	0.2067	4.472	119923	5.079
9	0.6900	2858.91	0.2067	4.472	229566	5.361
10	0.6900	3081.64	0.2067	4.472	490402	5.691
11	0.6900	2675	0.1296	4.472	103503	5.015
12	0.6900	2675	0.1645	4.472	111552	5.047
13	0.6900	2675	0.2041	4.472	120721	5.082
14	0.6900	2675	0.2532	4.472	132183	5.121
15	0.6900	2675	0.3215	4.472	148289	5.171

Table E-18 MDRM analysis results for Tests #22-23.

Moments	Entropy	i	0	1	2	3	4
m=1	1.8564	λ_i	1.795579	2.2349E-12			
		α_i		14.6493			
m=2	-0.1623	λ_i	543.9588	-303.2745	15.0621		
		α_i		0.6168	1.8022		
m=3	-0.1488	λ_i	1.19E+02	1.0594	0.2890	-19.7372	
		α_i		-0.2159	3.4803	1.4334	
m=4	-0.1506	λ_i	153.7723	-33.6849	0.7461	9.4019	-1.6392
		α_i		1.2621	3.0576	-2.3124	-2.1264

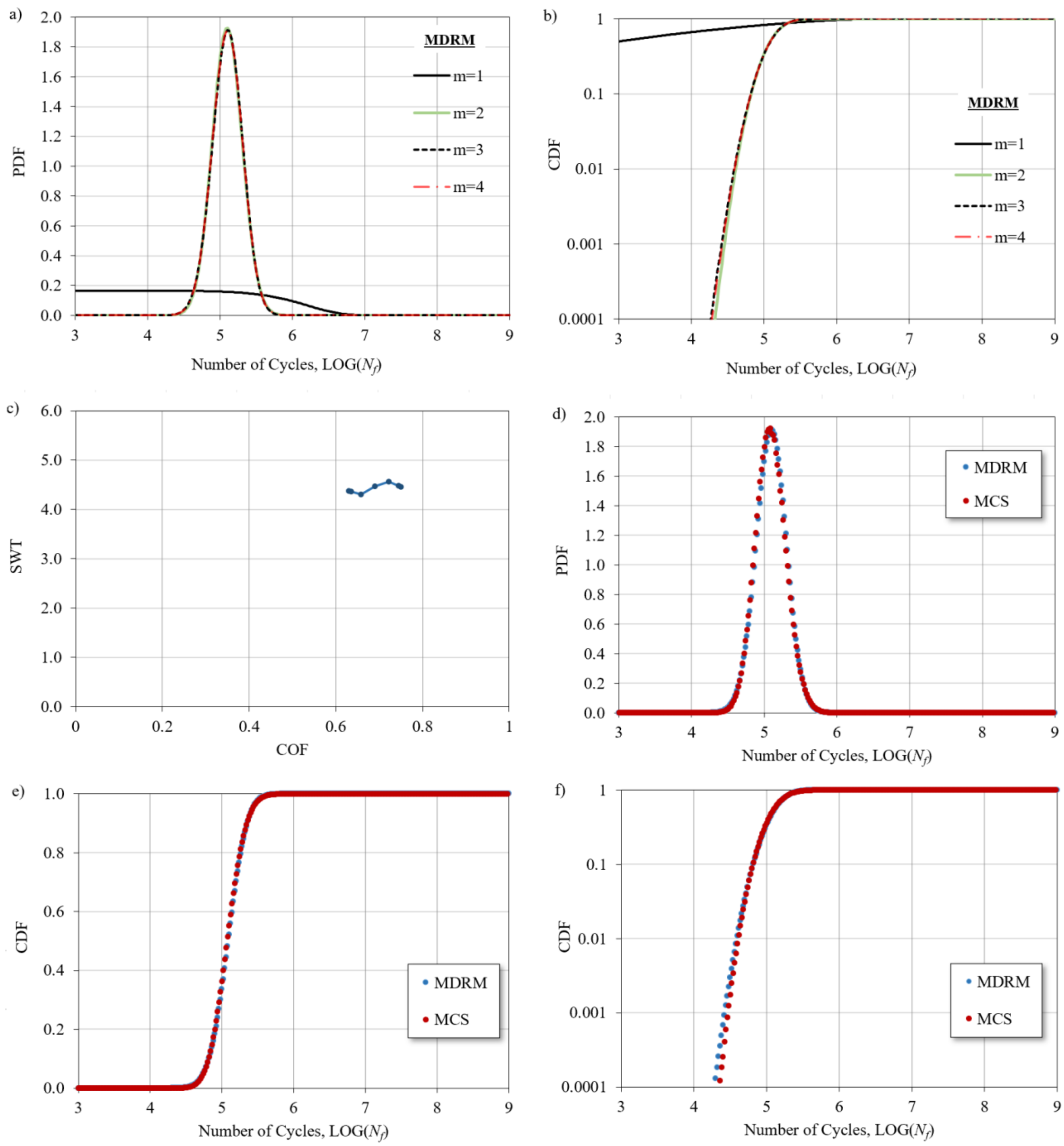


Figure E-9 Probabilistic analysis for Tests #22-23: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale (f).

Table E-19 Input gird for MDRM analysis of Tests #24-25.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.3402	2675	0.2067	2.847	1240877	6.094
2	0.4192	2675	0.2067	3.560	380784	5.581
3	0.5350	2675	0.2067	4.217	161766	5.209
4	0.6508	2675	0.2067	4.439	125776	5.100
5	0.7298	2675	0.2067	4.474	121029	5.083
6	0.5350	2316.23	0.2067	4.217	45096	4.654
7	0.5350	2496.68	0.2067	4.217	85628	4.933
8	0.5350	2671.66	0.2067	4.217	159856	5.204
9	0.5350	2858.91	0.2067	4.217	310039	5.491
10	0.5350	3081.64	0.2067	4.217	670210	5.826
11	0.5350	2675	0.1296	4.217	140100	5.146
12	0.5350	2675	0.1645	4.217	149891	5.176
13	0.5350	2675	0.2041	4.217	161033	5.207
14	0.5350	2675	0.2532	4.217	174940	5.243
15	0.5350	2675	0.3215	4.217	194450	5.289

Table E-20 MDRM analysis results for Tests #24-25.

Moments	Entropy	i	0	1	2	3	4
m=1	1.9484	λ_i	1.874457	1.2792E-11			
		α_i		13.1054			
m=2	0.4605	λ_i	335.6259	-240.6697	15.6567		
		α_i		0.4205	1.3534		
m=3	0.4532	λ_i	4.94E+02	63.7823	-287.6048	-138.8152	
		α_i		0.9871	0.3595	0.4641	
m=4	0.4578	λ_i	88.39245	25.2820	-98.0799	31.1974	437.0195
		α_i		1.2103	0.6771	-0.3850	-2.3819

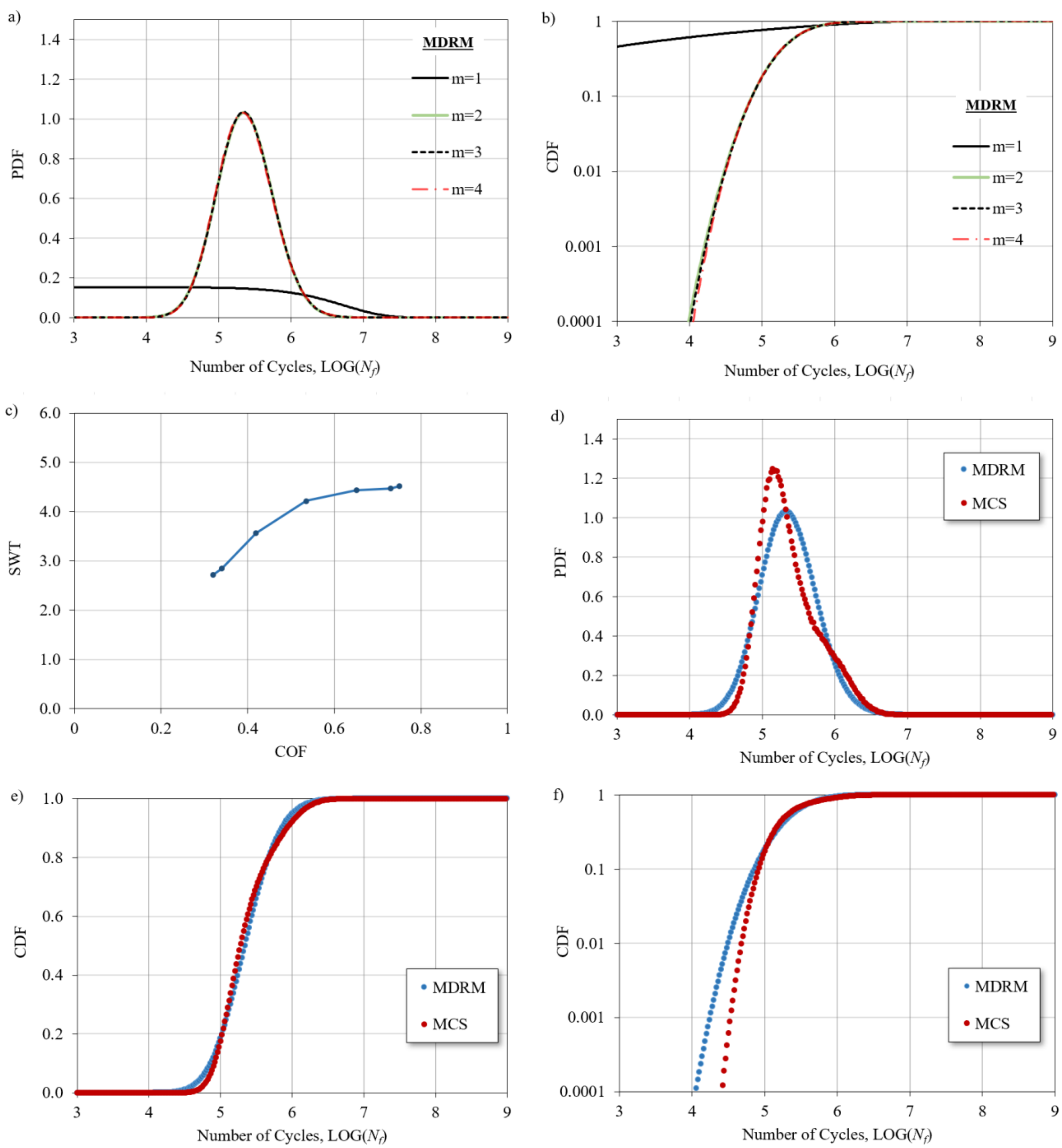


Figure E-10 Probabilistic analysis for Tests #24-25: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale (f).

Table E-21 Input gird for MDRM analysis of Test #26.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.3402	2675	0.2067	3.484	426095	5.630
2	0.4192	2675	0.2067	4.201	164777	5.217
3	0.5350	2675	0.2067	5.065	67101	4.827
4	0.6508	2675	0.2067	5.397	50064	4.700
5	0.7298	2675	0.2067	5.486	46488	4.667
6	0.5350	2316.23	0.2067	5.065	20794	4.318
7	0.5350	2496.68	0.2067	5.065	37309	4.572
8	0.5350	2671.66	0.2067	5.065	66365	4.822
9	0.5350	2858.91	0.2067	5.065	123274	5.091
10	0.5350	3081.64	0.2067	5.065	256027	5.408
11	0.5350	2675	0.1296	5.065	55286	4.743
12	0.5350	2675	0.1645	5.065	60613	4.783
13	0.5350	2675	0.2041	5.065	66700	4.824
14	0.5350	2675	0.2532	5.065	74334	4.871
15	0.5350	2675	0.3215	5.065	85112	4.930

Table E-22 MDRM analysis results for Test #26.

Moments	Entropy	i	0	1	2	3	4
m=1	1.8590	λ_i	1.795038	1.4806E-12			
		α_i		14.8826			
m=2	0.4194	λ_i	286.262	-222.2800	34.3741		
		α_i		0.5248	1.1837		
m=3	0.4147	λ_i	4.16E+02	33.8210	-211.8186	-128.6257	
		α_i		1.1369	0.3616	0.4064	
m=4	0.4048	λ_i	705.555	-266.7742	-466.7660	153.8642	130.3989
		α_i		0.3924	0.2779	0.7639	-11.2035

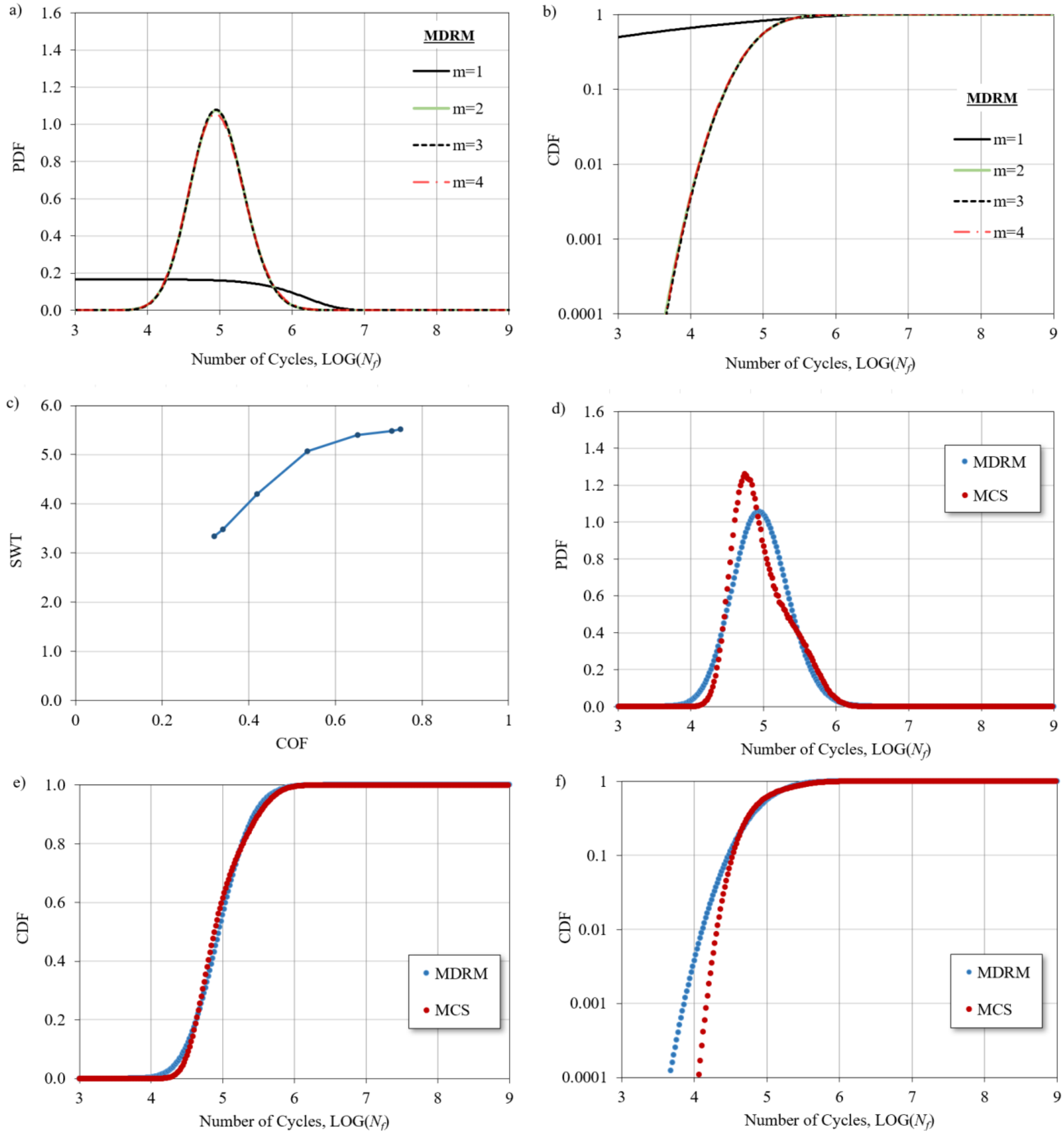


Figure E-11 Probabilistic analysis for Tests #26: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale (f).

Table E-23 Input gird for MDRM analysis of Test #27.

Variable	COF (X1)	σ_f' (X2)	ε_f' (X3)	SWT	N_f	Log(N_f)
1	0.2639	2675	0.2067	3.019	906483	5.957
2	0.3577	2675	0.2067	3.838	259061	5.413
3	0.4950	2675	0.2067	5.213	58720	4.769
4	0.6323	2675	0.2067	5.777	36886	4.567
5	0.7261	2675	0.2067	5.766	37178	4.570
6	0.4950	2316.23	0.2067	5.213	18510	4.267
7	0.4950	2496.68	0.2067	5.213	32914	4.517
8	0.4950	2671.66	0.2067	5.213	58084	4.764
9	0.4950	2858.91	0.2067	5.213	107100	5.030
10	0.4950	3081.64	0.2067	5.213	220868	5.344
11	0.4950	2675	0.1296	5.213	47970	4.681
12	0.4950	2675	0.1645	5.213	52814	4.723
13	0.4950	2675	0.2041	5.213	58354	4.766
14	0.4950	2675	0.2532	5.213	65310	4.815
15	0.4950	2675	0.3215	5.213	75141	4.876

Table E-24 MDRM analysis results for Test #27.

Moments	Entropy	i	0	1	2	3	4
m=1	1.9064	λ_i	1.83135	9.491E-12			
		α_i		13.5758			
m=2	0.7434	λ_i	221.5865	-185.5604	22.1133		
		α_i		0.3895	1.0811		
m=3	0.6892	λ_i	6.44E+02	1233.2783	-1904.5363	2171.5240	
		α_i		-0.2392	-0.1527	-3.6127	
m=4	0.7095	λ_i	117.7668	-8.0449	773.9425	302.1131	-635.7065
		α_i		-0.7401	-1.8366	-0.3337	-0.4037

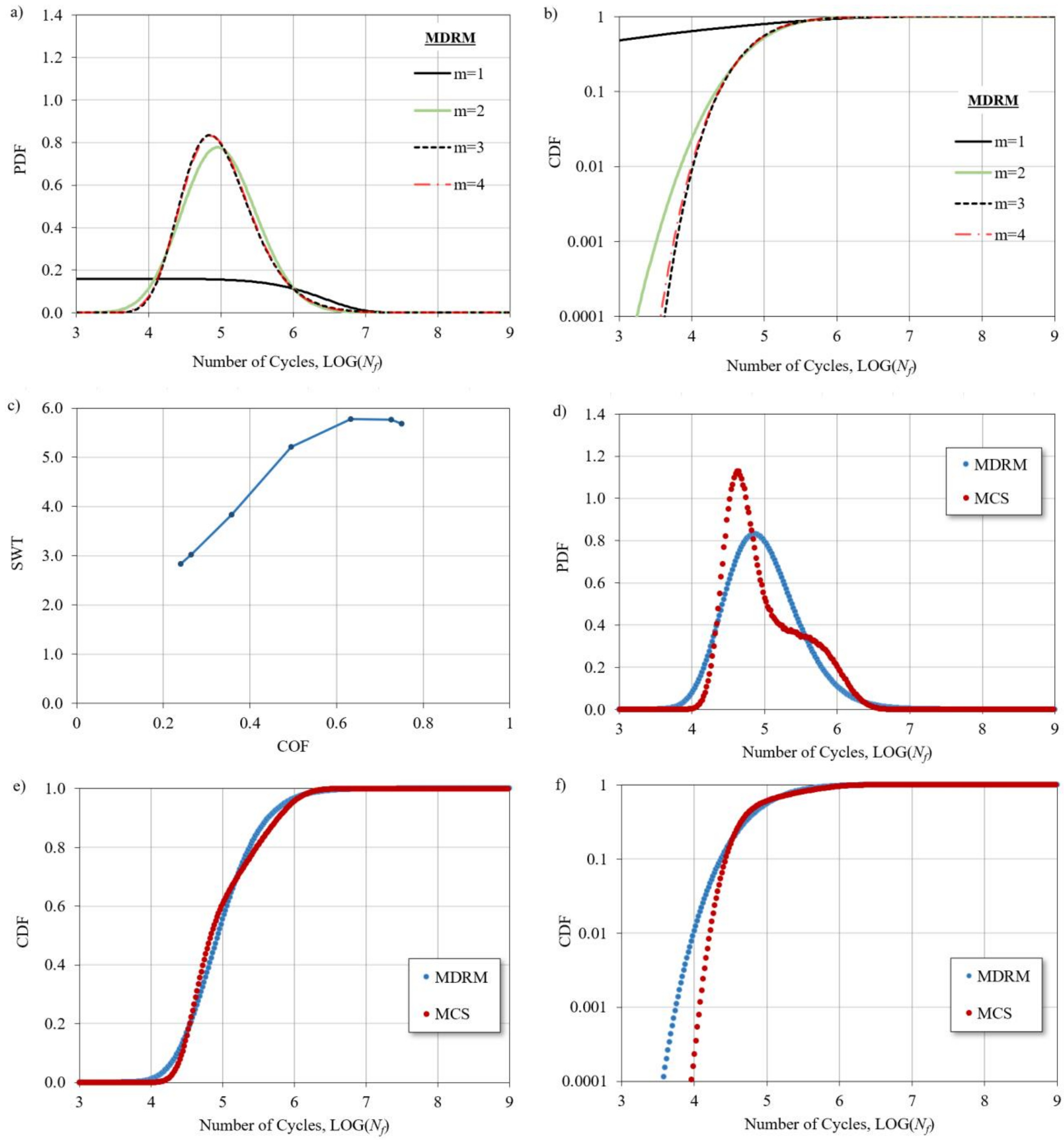


Figure E-12 Probabilistic analysis for Test #27: PDFs (a) and CDFs (b) based on different terms in MDRM analysis; SWT-COF map (c); Comparing MDRM and MCS resulted PDFs (d) and CDFs (e), and comparing MDRM and MCS results for CDF on logarithmic scale (f).