Shaft Crack Monitoring via Torsional Vibration Analysis; Part 1 – Laboratory Tests

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ABSTRACT

Torsional vibration signature analysis has shown the potential to detect shaft cracks during normal machinery operations of rotating equipment. The method tracks characteristic changes in the natural torsional vibration frequencies that are associated with shaft crack propagation. The method is generally applicable to many types of rotating equipment. A laboratory scale rotor test bed was developed to investigate shaft cracking detection techniques under controlled conditions. A sample shaft was seeded with a semi-elliptical surface crack, which was propagated in three point bending. The fatigue crack was incrementally grown in nine steps, with depths ranging from approximately 0 – 60% of the shaft diameter. After the crack was grown to each pre-defined depth, the shaft was installed in the rotor test bed and the changes in shaft torsional vibration features observed. The first torsional natural frequency is shown to be sensitive to the shaft crack depth, which for the crack depths tested produced a 2 Hz frequency drop. The relationship between crack depth and torsional natural frequency is nonlinear. The test data show that changes in the torsional shaft frequency in the range of 0.1 to 0.2 Hz. can be detected by a visual inspection. This study points to the potential of using online torsional signature analysis as a diagnostic for shaft crack monitoring in rotating equipment.

INTRODUCTION

The importance of shaft crack detection in nuclear power plants is apparent when considering the impact of past failures. For example, Primary Coolant Pumps (PCPs) have experienced shaft cracking and subsequent failure, often with little or no warning from state-of-the-art crack detection systems. The financial loss associated with a forced outage caused by such a shaft failure is substantial. Recently, pre-1974 Westinghouse Reactor Coolant Pumps (RCPs) have come under particular scrutiny, as at least five have experienced significant cracking. A root cause analysis indicated that Model 93A pumps which are operated in counterclockwise flow loops are especially susceptible to developing shaft cracks [1].

In late 2000, Tennessee Valley Authority (TVA) Sequoyah Unit 1, RCP 4 experienced severe cracking that resulted in an extended forced outage. The crack detection system on the RCP provided no warning of crack propagation. After shutdown, inspection revealed a circumferential crack of 252º, and only one-third of the cross-sectional area remaining.

The unexpected loss of a Steam Generator Feed Pump (SGFP) in a Pressurized Water Reactor (PWR) or a Reactor Feed Pump (RFP) in a Boiling Water Reactors (BWR) often results in a unit trip and subsequent operation at reduced power. For instance, the loss of a SGFP due to turbine shaft cracking occurred at Plant Vogtle (Southern Nuclear) in the mid-90s. The result was a unit trip, and a 3-4 day outage until operation was resumed. In addition, the unit operated at reduced power (~70%) for approximately one week to facilitate turbine repair resulting in a substantial monetary loss.

In addition, PCPs in BWRs have had shaft-cracking problems. Recently, for instance, Plant Hatch (Southern Nuclear) has replaced several PCPs due to potential shaft cracking problems identified by the vendor. Other pumps that have experience shaft cracking include Condensate Pumps and Centrifugal Charging Pumps (20 shaft failures of current designs of PWRs used in the USA). Although the failure of these pumps due to shaft cracking does not generally result in unit trips or reduced power operation, the unexpected failure can create maintenance scheduling problems and increased safety risk.

A number of shaft failures have been also occurred in non-nuclear electrical generating facilities. For example, the hydro-turbine drive shaft on Southern Company's Morgan Falls Unit 5 completely fractured [2].

These mechanical failures have motivated efforts to develop condition based monitoring methods for the structural integrity of rotating equipment components. One such approach utilizes torsional vibration signature analysis. As the integrity of rotating components degrades due to the initiation and propagation of fatigue cracks the torsional natural frequencies decrease. This method appears to be less sensitive to changes in the pump rotor context (e.g., seals, oil film, and supports) than the existing crack detection systems. Therefore, the tracking of torsional natural frequencies can be used as a health diagnostic feature in rotating machinery.

The purpose of this work is to explore the ability of torsional vibration signature analysis as a diagnostic to monitor the development and growth of fatigue cracks in rotating drive shafts. A series of laboratory tests were performed during the investigation. Initially a small notch was machined in a test shaft to produce a stress concentration to initiate a fatigue crack. The shaft was then placed in a three point bending apparatus and cyclically loaded to grow a fatigue crack. Crack size was measured ultrasonically and from the beach marks on the fracture surface. The shaft was installed in a torsional test rig and operated to characterize the torsional vibration signature. The shaft was then subjected to additional fatigue cycling and the inspection process repeated. The torsional vibration signature was examined at each fatigue increment to examine the ability and sensitivity to detect and track growth of the fatigue crack.

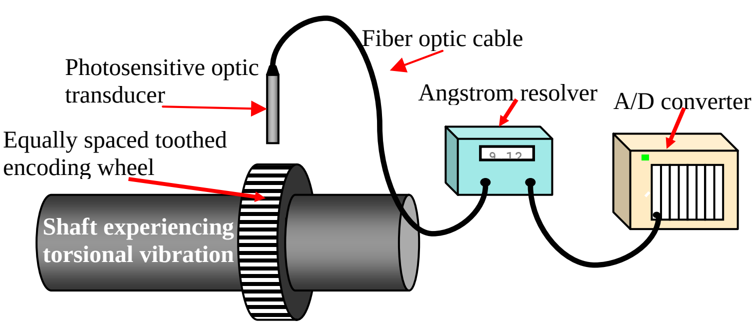

Figure 1. Torsional vibration measurement instrumentation.

TORSIONAL VIBRATION ANALYSIS AS A ROTATING MACHINERY DIAGNOSTIC

A system to measure torsional vibration of a rotating shafting system is shown schematically in Figure 1. Signal detection involves four main aspects, shaft encoding, transduction, data discretization and demodulation. The shaft encoding system can use a variety of approaches including a timing gear or optical encoder. Depending on the shaft encoding device a number of transducers are viable, including infrared intensity based reflective fiber optic sensors and Hall Effect transducers.

Figure 2. Shaft specimen in three point bending MTS fatigue machine.

The demodulation can be performed either by an analog or digital process. This work implements the digital method called the TIMS (time interval measurement system) [3]. First, a high speed timer circuit (in the MHz range) is used to record the passing times of each line on the encoding device. The passage times are then converted to angular shaft velocity. Various investigators have addressed a number of issues and proposed enhancements to this approach [4-8]. These enhancements allow for correction of transducer imperfections and the removal of fixed order components. Many of these refinements are implemented in the torsional vibration measurement system used in this effort.

FATIGUE CRACK GROWTH

The test specimen was a 0.625 inch diameter ANSI 316 stainless steel shaft. To initiate a crack, a 0.010 inch deep notch was machined into the midlength of the shaft specimen with the edge of a milling machine tool following ASTM E399 standards for fatigue precracks. The specimen was mounted on an MTS 642.10B 3-point bend fixture and cyclic loading was applied with a servohydraulic MTS 810 test stand operated by a MTS 458.20 electronic controller. The fatigue test hardware is shown in Figure 2. Tests were performed to determine the cyclic loading specifications needed to grow the crack to a predefined depth without causing permanent deformation. This load and cycle information was used to control the crack growth.

Upon completion of a cyclic loading sequence, the fatigue crack was quantified using both a visual and an ultrasonic non-destructive evaluation method. The visual inspection entailed viewing the crack with a telescope and measuring the fatigue crack surface length while the shaft was under a constant lateral load in the 3-point bend fixture. Two nondestructive evaluation procedures using ultrasonic waves were applied to map the crack front. One method used a conventional ultrasonic inspection technique with longitudinal waves to estimate the uncracked shaft cross section. The second method used a novel wedge transducer that emits both surface and

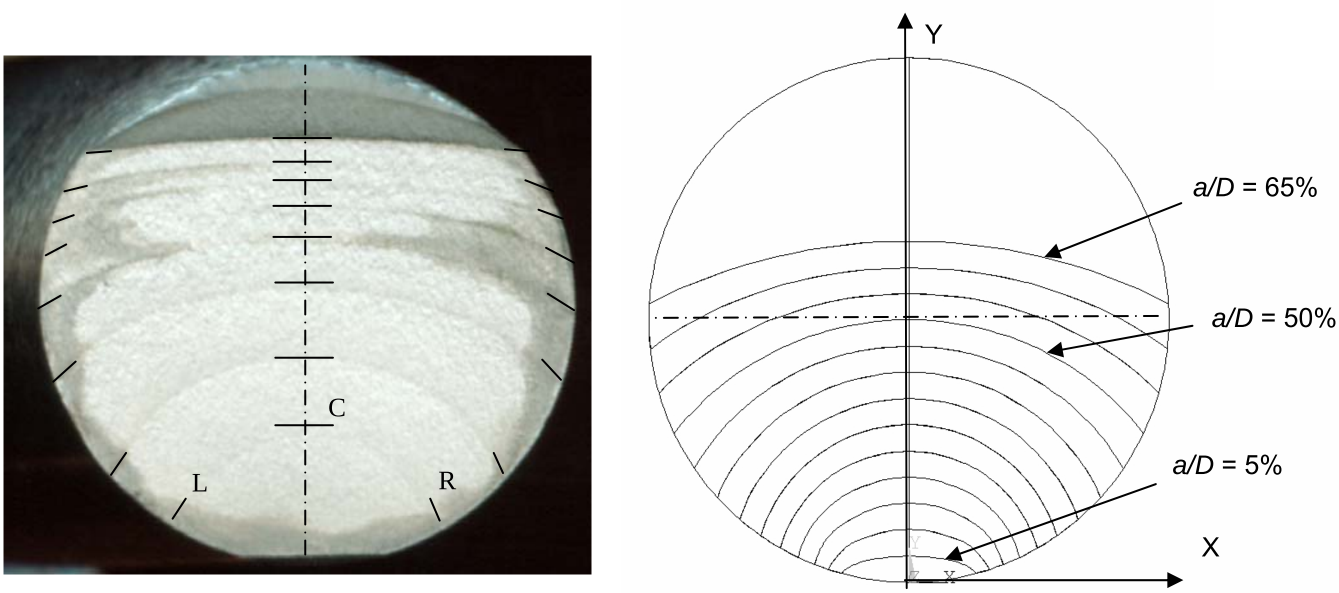

Figure 3. Shaft section beachmarks.

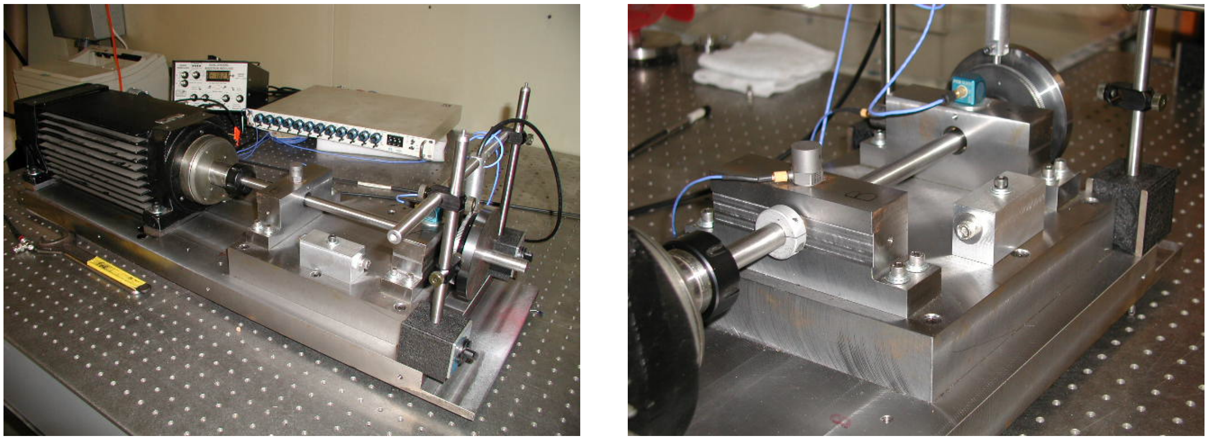

Figure 4. Laboratory torsional vibration test rig.

shear waves.

At the completion of the fatigue cycling a destructive crack evaluation was performed. The shaft was sectioned and the cracked cross-section viewed with a metallograph to help correlate the NDE results with actual crack characteristics as seen in Figure 3. The crack metric (a) is expressed as the distance from the shaft surface to the extreme location of elliptical crack front along a diametric line.

After each fatigue cycling increment, an MTS 319 Axial-Torsion test machine was used to determine the reduction in torsional rigidity, GJ, relative to the crack condition. Two fully reversed cycles of a triangular torque waveform were applied at a rate of 20 lb-in/s and torque and angle of twist were measured. Linear regression was employed to determine torque-angle relationship and the torsional rigidity calculated.

TORSIONAL VIBRATION TEST STAND

A test rig was constructed to measure the torsional vibration of the cracked rotating shaft under controlled laboratory conditions and is shown in Figure 4. The shaft specimen was mounted in Rulon-J non-metallic flanged sleeve bearings and rotated by a variable speed DC Perske motor.

An encoding wheel was constructed to sense the torsional vibration with an outer diameter of 3.575 inches and 180 teeth with a 0.125 inch depth. An infrared fiber optic intensity reflective transducer was used to sense the wheel tooth passage. The tooth passage times were sensed and recorded with a National Instruments PCI-6602 Timer/Counter Board using an 80MHz clock reference.

Once recorded, the encoding tooth passage times were processed with a specialized torsional vibration algorithm based on the Time Interval Measurement System method. The spectrum was estimated using the parameters listed in Table 1. Of particular note is the high spectral resolution obtained by the large FFT block size and the steps taken to enhance the data quality by applying an encoder correction scheme and removal of any order content [9].

Processing Selection	Specification
Analysis frequency range, $f_{analysis}$	0 –2048 Hz
FFT block size, N	131,072
Number of spectral lines	65,536
Spectral resolution, Δf	0.03125 Hz
Encoder correction routine	Implemented [9]
Order removal routine	Implemented [9]
Constant time step resampling Order-to-Frequency domain routine	Implemented [9]
FFT spectral estimation data window	Hann
Ensemble averages	60

Table 1. Torsional spectra processing parameters

A typical torsional spectrum is shown in Figure 5. The spectrum tends to have some low level harmonic content and a single high amplitude spectral line around

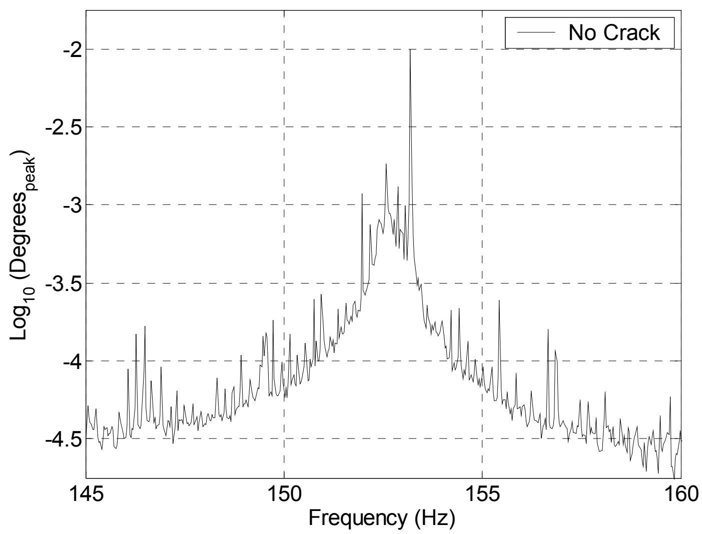

Figure 5. Typical torsional vibration spectrum.

152 Hz. This spectral content was discovered to be associated with artifacts induced by the motor speed controller. The underlying shape of the spectrum shows a classical under-damped modal response indicative of

the first torsional natural of the system with nominal operating conditions. The torsional natural frequency for each test was determined by a visual peak-picking method. Subsequent torsional vibration testing will track the change of this natural frequency as the fatigue crack is propagated.

TORSIONAL RESPONSE OF A SHAFT WITH A PROGRESSIVELY GROWN FATIGUE CRACK

A test sequence was performed that evaluated the torsional vibration signature of a shaft with a fatigue crack grown progressively to nine different stages. The crack was located one half the axial distance along the test shaft seen in Figure 4. The following test sequence was performed.

1. A new shaft, with a milled notch, was tested in the torsional test rig to establish the baseline signature.
2. The fatigue crack was grown as closely as possible to the predefined schedule of depths.
3. The crack was nondestructively inspected and the torsional rigidity tested.
4. The shaft was tested in the torsional test rig to determine the torsional natural frequency. The vibration signature was measured with eight to ten complete disassembles and reassembles of the shaft in the torsional test stands. The multiple tests were necessary to statistically establish the system natural frequency due to variability induced by the assembly-disassembly of the shaft into the torsional test stand.
5. Steps 2-4 were repeated eight times.
6. The shaft was destructively sectioned to evaluate the crack fronts at each torsional vibration determination.

Shifts in the torsional natural were observed as a result of the mechanical disassembly-reassembly of the shaft from the test stand. The variation was typically within a range of $\pm$ 0.2 Hz. This variation was judged too large, since expected shifts due to an incremental growth of the shaft fatigue crack were expected to be similar in magnitude. Minitab [10] was used to identify outliers based upon a statistical analysis of the natural frequencies from the repetitive test runs for each crack depth. A box plot showing frequency versus fatigue crack step number was created in Minitab, shown in Figure 6. The bottom of the box is the first quartile of data (Q1), and the top of the box is the third quartile of data (Q3). Thus, the box accounts for the middle 50% of the data. Within the box, mean frequency values are denoted with a solid dot. The line across the box is the median.

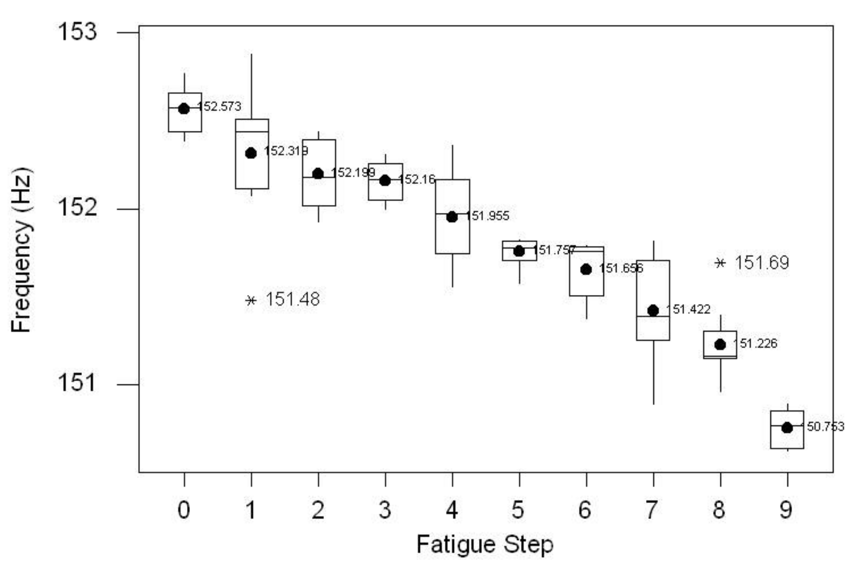

Figure 6. First torsional natural frequency vs fatigue crack depths.

The whiskers are lines that extend from the top and bottom of the box. The length is determined to be the values that are within 1.5 times the size of the box. All outliers were eliminated and the mean of the remaining data points calculated and used to represent the identified natural frequency for each test.

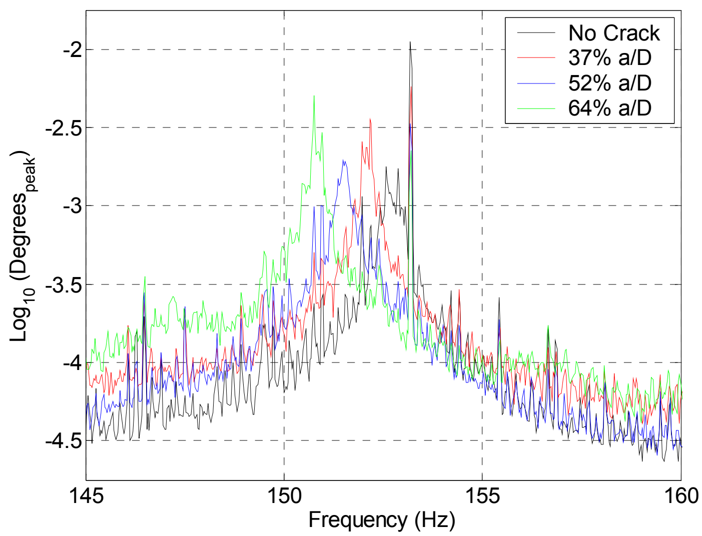

Figure 7. Torsional vibration spectrum from the laboratory test rig with the shaft in four different fatigue crack depths.

The torsional vibration spectra around the first natural frequency for four selected tests increments (crack depth to shaft diameter ratios a/D = 0%, 37%, 52%, 64%) that span the test range are shown in Figure 7. A reduction in system natural frequency as the crack grows is apparent. The mean natural frequency, after the elimination of any outliers, at each fatigue crack depth is shown in Figure 8. The results show that there is a gradual decrease in the shaft natural frequency with approximately a 2 Hz drop observed for a crack depth of a/D = 64%.

The sensitivity of the torsional natural frequency shift to crack growth depends on the inherent torsional system dynamics in relation to the spatial crack location. Therefore, certain modes show more sensitivity to natural frequency shifts for a crack in a specific location. Rotor dynamic finite element modeling has proven effective in developing an understanding between the spatial crack formation and the respective modal frequency sensitivities. Therefore, the correlation of the torsional natural shift to crack depth is very system specific and cannot be generalized. However, based upon the testing performed an assessment of the ability to detect a shift in natural in frequency can be made. An inspection of the torsional spectra from the laboratory test rig showed that a nominal detectable shift in the frequency by a visual inspection was in the range of 0.1 to 0.2 Hz.

SUMMARY

The torsional vibration tests with the progressively grown fatigue crack showed:

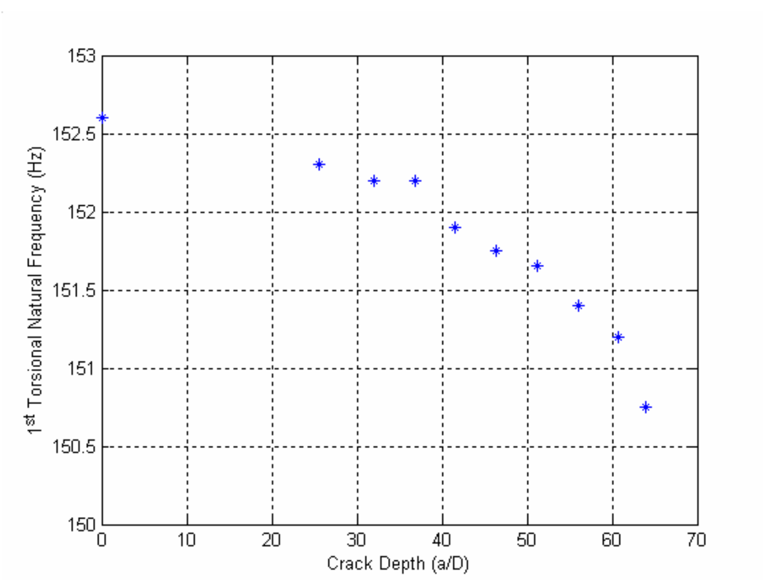

Figure 8. First torsional natural frequency vs fatigue crack depths.

1. There is an identifiable natural frequency change that can be tracked in relation to the seeded fault condition of the shaft.
2. The torsional rigidity (GJ) showed a measurable decrease in relation to the crack growth.
3. A decrease in the first torsional natural frequency was observed to be nonlinear with respect to the diametric crack depth.
4. Changes in natural frequency in the range of 0.1-0.2 Hz were identifiable by a visual inspection of the spectrum.

The torsional vibration based monitoring method demonstrated the ability to reliably detect natural frequency shifts in the range of 0.1 - 0.2 Hz. This frequency shift is within the range of frequency shifts caused by shaft cracks, and hence shows the potential to enable online diagnostics and subsequent prevention of shaft failure due to crack growth by utilizing life prediction modeling. Furthermore, other processing approaches can be applied to further refine the ability to track small crack growth. More accurate natural frequency identification methods should be applied to the data beyond the visual inspection. The estimation and trending of the modal damping as the crack grows may also prove useful. The correlation between the crack depth and the measured natural frequency shift requires attention. System torsional dynamics along with incorporation of effective methods to model the localized stiffness change of a fatigue crack must be used. Only then can the torsional diagnostic be used to estimate the Remaining Useful Life (RUL) of a shaft while in service.

ACKNOWLEDGEMENTS

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