ENGINE CRANKSHAFT FAILURES DUE TO TORSIONAL NATURAL FREQUENCY EXCITED BY DUAL FUEL OPERATION

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Abstract

Two diesel engine–generator systems were originally built and certified for emergency power at a nuclear power plant. Now these systems are repurposed for continuous power generation in South America. The engines were converted by the OEM to operate on dual fuel due to reduced energy cost of natural gas using the OEM's obsolescent dual fuel technology. Two failures of the engine crankshafts have occurred after the conversion. In both instances, spiral cracks occurred in the engine crankshaft between throws 7 and 8 (flywheel end), which is indicative of high torsional vibration. Through torsional analysis and field measurements, the root cause for the crankshaft failures was found to be excitation of the first torsional natural frequency (TNF) by the 4th engine order. When operating on diesel fuel, for which the units were originally designed and tested, the 4th engine order is low amplitude and stable. However, when operating the engine in dual fuel mode, the amplitude of the 4th order varies considerably and can be quite high. Measured pressure traces in the engine cylinders showed increased variation with dual fuel versus diesel. The difference in excitation created by dual fuel versus diesel combustion, coupled with the generally stochastic nature of gaseous combustion, results in excitation at the 4th engine order that had not been previously recognized. Recommendations are made to improve the reliability of the engine operating with dual fuel.

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INTRODUCTION AND BACKGROUND

The two engine-generator units discussed in this paper were originally provided to a nuclear generating station in the United States. Years later, the units were moved to South America. In approximately 2010, the engines were converted from diesel to dual fuel* operation by the OEM using outdated technology. Dual fuel (DF) conversion was done to reduce the operating cost. In DF mode, air and natural gas is drawn into the cylinder, with a lean air-to-fuel ratio. A small amount of diesel is injected (pilot fuel) and auto-ignites near the end of the compression stroke causing the natural gas to burn. Table 1 provides details on the two engine-generator units that are shown in Figure 1.

Table 1. Equipment Description

Engine	Enterprise Model DSRV-16-4 Articulated Rod Design, No Damper Four quarter-round rotating counterweights on crankshaft. Unit A – no micropilot system. Unit B – retrofitted with a micropilot system after crankshaft failure.
Flywheel	Outside Diameter = 2300 mm (90.6-inch)
Generator	General Electric, Rated 7000 kW at 450 RPM

Figure 1. Engine – generator units

The 17-inch (430 mm) bore Enterprise R/RV-4 series engine was first introduced in 1968 as a four-stroke diesel, dual fuel and heavy fuel engine in L-6 and L-8 and V-12, V16 and V-20 configurations. Over 300 of these engines were built with an ultimate rating of 228 psi BMEP at 450 RPM. The V version

* For the purposes of this paper, "dual fuel" means a compression ignition cycle in which 5% to 6% liquid pilot fuel delivered by the conventional mechanical injection system is used to ignite a lean pre-mixed charge of air and fuel. Micropilot means less than 2% pilot delivered by a dedicated system.

utilizes an offset articulated rod (Figure 2) to minimize overall engine length. As a result, the articulated rod (right) bank does not exhibit pure slider crank geometry resulting in potential 4th order excitation on the V-16 model engine.

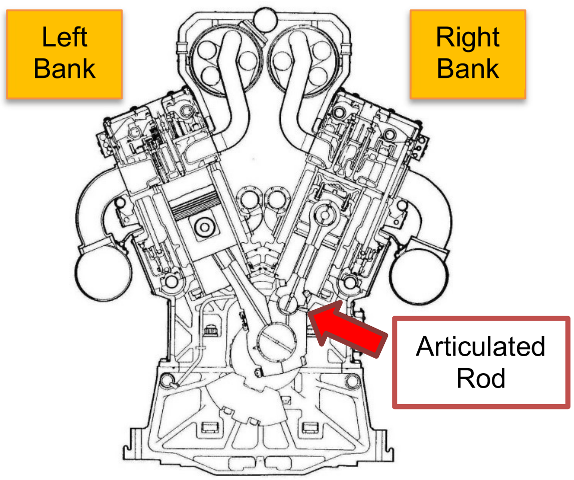

Figure 2. Cross section of RV-4 engine

Prior to DF conversion, the engines had accumulated ~2,000 hours each. Shortly after conversion, the engine crankshaft on Unit B failed after 1,300 hours of operation in the DF mode. The engine crankshaft on Unit A failed after accumulating 8,700 hours of operation in the DF mode. Both failures occurred in the main bearing section of the engine crankshaft between throws 7 and 8 as shown in Figure 3.

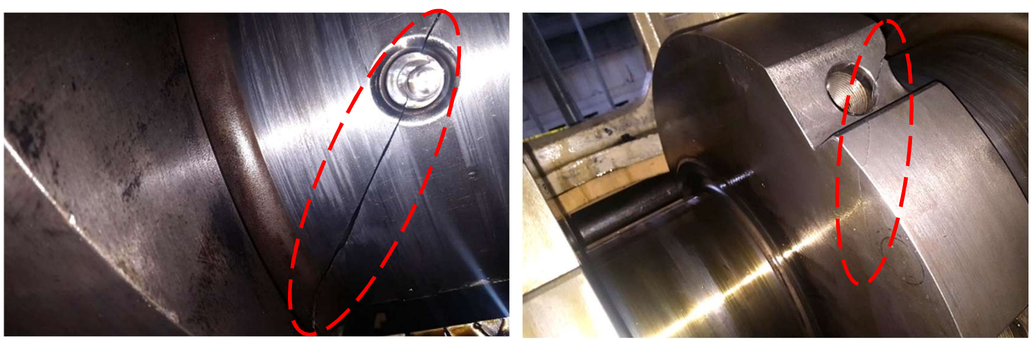

Figure 3. Photos of cracks in engine crankshafts

A crack occurring at a 45-degree angle to the shaft rotation is typical of a system experiencing high torsional vibration [2]. Cracks will normally start at a stress riser such as oil hole and then propagate through the crankshaft web, etc. The cracks were typical of high-cycle fatigue failure normally associated with ~10 million cycles (~700 hours of operation). While the engines, particularly Unit A, had operated longer than this since conversion, the life in the diesel mode prior to conversion strongly suggested some intermittent operating condition in the dual fuel mode was the root cause of the problem.

VERIFICATION OF TORSIONAL NATURAL FREQUENCIES

The TNFs were measured using an HBM torsiograph mounted on the end of the engine crankshaft as shown in Figure 4. Based on previous experience [3], the TNFs needed to be verified to determine if the rotating counterweights on the engine crankshafts should be adjusted. The counterweights as installed were found to be optimum. It was also confirmed that the measured TNFs matched the results from the OEM torsional analysis. Therefore, no changes to the existing counterweights were recommended.

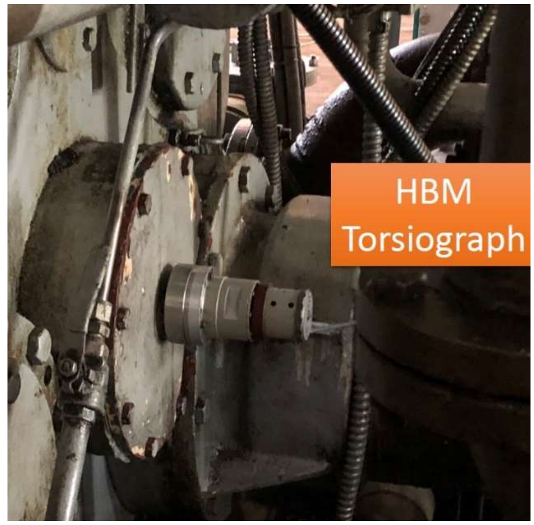

Figure 4. Measurement location

First TNF

Multiple shutdowns were captured. The ramp rate is slow enough that waterfall plots could be created as shown in Figure 5. By tracking the engine orders, the TNFs could be verified. As shown in Figure 6, the first TNF was 28.8 – 29.0 Hz. The torsional mode had an amplification factor (AF) of 32. Note that this engine crankshaft does not have a torsional damper.

Second TNF

The second TNF is more difficult to detect using the HBM torsiograph mounted on the front of the engine. The second TNF is approximately 56.0 – 56.5 Hz. This is near 7.5× engine speed and 5% from 60 Hz electrical line frequency.

Comparison with OEM Torsional Analysis

As shown in Table 2, both units have similar TNFs that agree well with the OEM torsional analysis and verification testing conducted in the 1970's. Since there was some separation from the 4th engine order, and the 60 Hz electrical line frequency, no changes to the rotating counterweights are recommended. However, the 4th engine order is still being amplified by approximately 15 times due to the proximity of the first TNF.

Table 2. Summary of TNFs

	Unit A	**Unit B**	**OEM**
First TNF	28.8 – 29.0 Hz	29.0 Hz	29.2 Hz
Second TNF	56.7 Hz	56.0 – 56.5 Hz	57.8 Hz

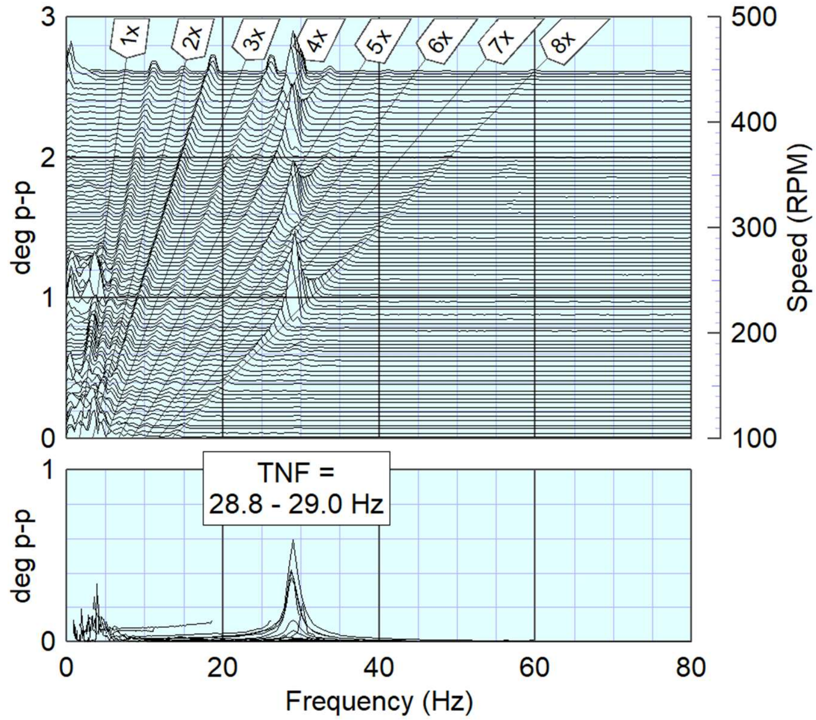

Figure 5. Waterfall plot taken during shutdown

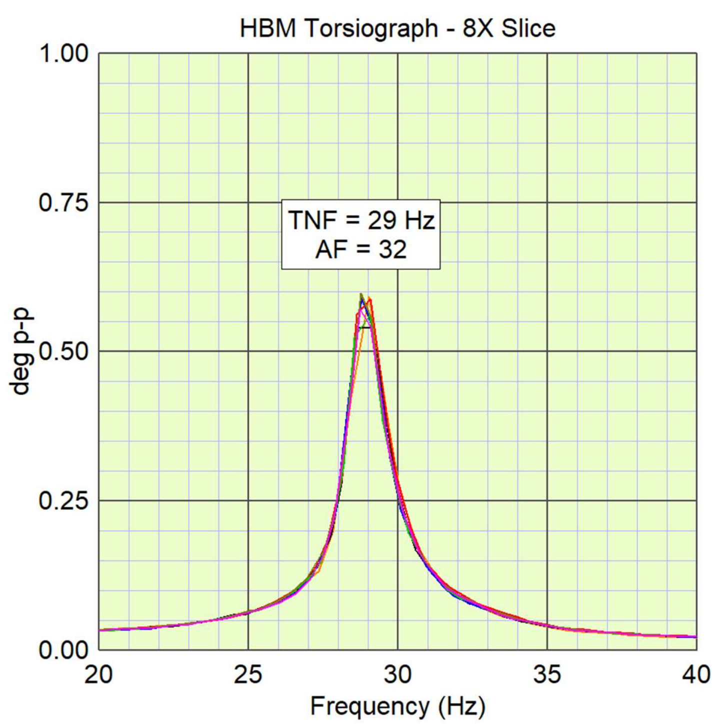

Figure 6. Order tracks from multiple runs

POTENTIAL IMPACTS OF DUAL FUEL OPERATION

All prior torsional analysis and testing of this engine type by the OEM had focused on diesel operation. Except for marine applications† such analyses assume all of the engine cylinders are equally balanced with no cycle-to-cycle variation. Prior experience with this engine type has shown bank-to-bank imbalance in diesel operation can occur due to mechanical failure issues, such as incorrectly adjusted fuel rack linkage on one bank or partially closed air damper on one bank. This could excite the 4th engine order resulting in a failure of the crankshaft.

Several failures had also occurred on dual fuel engines. However, the root cause(s) was never clearly defined, and no prior work had examined the potentially unique aspects of dual fuel operation such as:

- Cylinder-to-cylinder imbalance due to uneven fuel gas supply to each cylinder.
- Cycle-to-cycle variation due to inconsistencies in pilot ignition and flame propagation.
- Differences in the overall combustion trace between diesel and dual fuel resulting in different levels of torsional excitation.

Therefore, the field test program examined all three aspects in comparison to diesel operation (Figure 7). These test results confirmed that dual fuel operation had the potential to excite the 4th engine order for all three reasons.

Cylinder-to-Cylinder Imbalance

Simulation indicated (and testing confirmed) that various cylinder balance solutions could result in different 4th engine order responses. Optimum torsional response was obtained with the average of the peak pressures on the left bank (master rod side) 3.5% lower than the right bank (articulated rod side). Other balance solutions could significantly increase the 4th order response due to the articulated rod design.

Cycle-to-Cycle Instability

Testing confirmed that any engine parameter changes which reduced stability, such as reduced pilot fuel or leaner air/fuel ratio, significantly increased the 4th order torsional response. However, the variation revolution-to-revolution was substantial, with some very high 4th order responses interspersed with more typical levels. This indicated the stochastic nature of dual fuel combustion results in a different peak pressure “balance” solution for every revolution.

Some of these random solutions are little different than diesel operation where others, including misfires and over-pressure cycles, result in double or more the 4th order response exceeding the stress level for infinite crankshaft life. Figure 8 shows an example of pressure traces from 112 firing cycles that had cycle-to-cycle instability as well as cylinder-to-cylinder imbalance.

† Analyses for marine applications consider the worst-case impact of a non-firing cylinder.

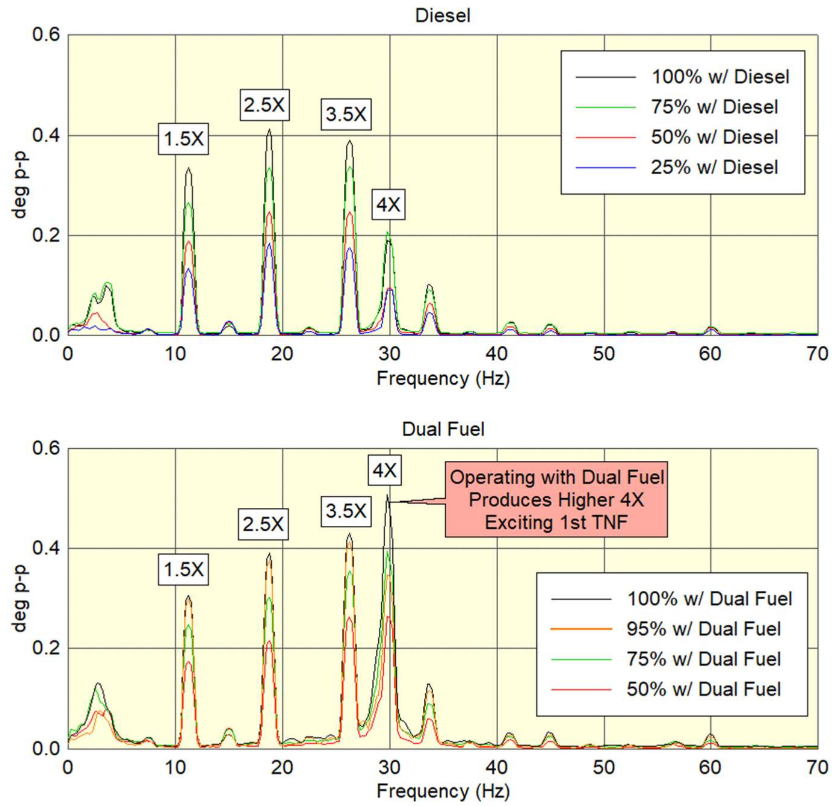

Figure 7. Comparison of torsional response in diesel and dual fuel modes

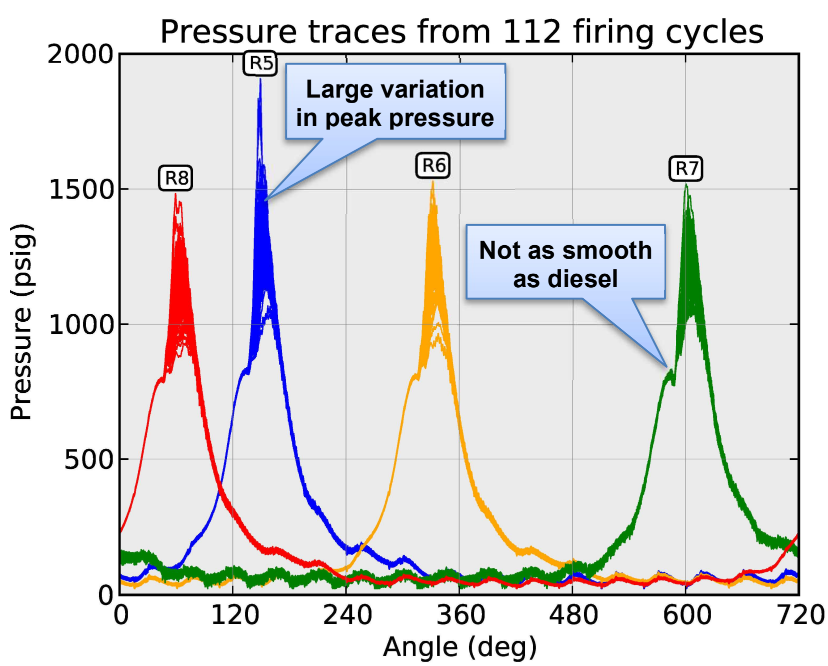

Figure 8. Engine firing cycles with dual fuel operation and 9 mm diesel rack position

Comparison of Pressure Traces (Diesel versus Dual Fuel and Micropilot)

The testing focused on Unit A utilizing a conventional diesel fuel injection system to deliver 5% to 6% pilot fuel. Unit B had been fitted with a dedicated micropilot system delivering 1% to 2% pilot fuel. As reflected in Figure 9, the diesel traces are significantly smoother than operation on dual fuel, which exhibits a relatively rapid rise is combustion pressure at top dead center (TDC).

Although the micropilot system on Unit B exhibited some instability it was generally more stable than the conventional dual fuel. Moreover, the micropilot pressure traces were smoother and closer to the typical shape for diesel [4] eliminating the rapid rate of pressure rise seen with the conventional dual fuel system. While beyond the scope of this work, subsequent analysis indicates this is due to the poor fuel injection consistency and poor fuel preparation of the conventional diesel injection system. While 5% to 6% pilot is injected, less than 1% triggers ignition with the remainder burning in near knock-like combustion near TDC [1] in this case contributing to excitation at the 4th engine order.

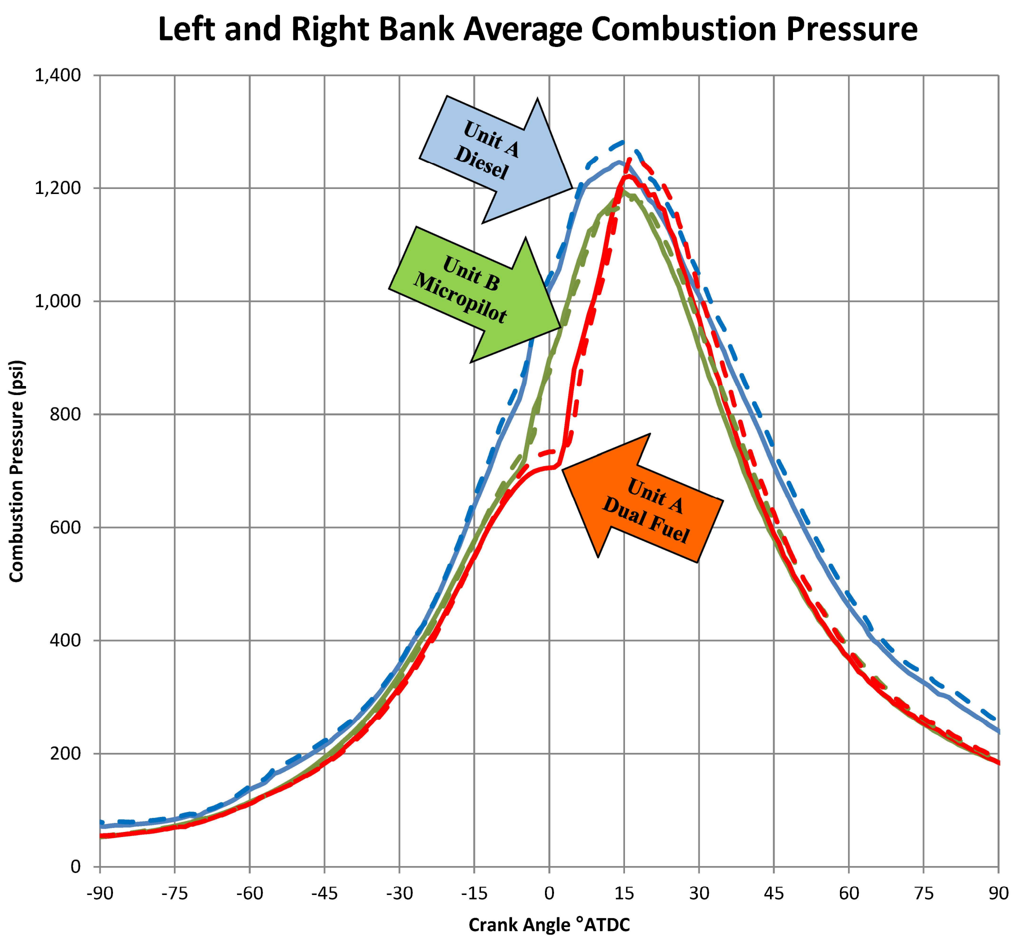

Figure 9. Comparison of pressure traces for diesel, dual fuel, and micropilot combustion

CONCLUSIONS AND RECOMMENDATIONS

These engines operated successfully with diesel fuel but suffered crankshaft failures after conversion to dual fuel operation. The failure was due to the unique characteristics of dual fuel combustion, including cylinder-to-cylinder imbalance, cyclic instability, and differences in combustion pressure. Intermittent excitation of the first TNF by the 4th engine order resulted in the accumulation of damaging cycles exceeding the infinite life factor of safety.

After testing, the units were left in "tuned" condition and had acceptable torsional vibration. Since the cylinder pressures could change over time, short-term (items 1-3) and long-term (items 4-5) recommendations were made:

1. Optimum rack position was determined in the current configuration. For the best stability, it was recommended not to reduce too much pilot fuel.
2. Adjust peak pressures on the left bank to be 30 - 40 psi lower than the right bank to reduce torsional vibration, particularly at 4× engine order.
3. Temporarily limit the generator power to 95%.
4. Install a permanent torsional vibration monitor to alert operators of a problem before any damaging events occur. The EDI torsional vibration monitoring system has been in use for over three years at the plant.
    a. Ideally, the torsional oscillation at 4× should be below the normal values for 1.5×, 2.5× and 3.5× engine orders and no greater than it was for diesel operation.
    b. Engine misfire can cause elevated torsional oscillation at 0.5× order. For smooth operation, the 0.5× order should be ≤ 0.1 deg p-p.
5. To improve the dual fuel pressure traces, reduce variation and excitation, and possibly reduce the amount of pilot fuel required:
    a. Install a micropilot system to smooth the dual fuel pressure traces (similar to diesel).
    b. Continuous pressure monitoring of all engine cylinders.
    c. Electronic port fuel gas injection to better regulate the natural gas.
    d. Retrofitting the existing crankshaft with a viscous damper was considered but deemed impractical. There was a damper option for marine applications where the engine operates over a speed range.

REFERENCES

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