# Rotating Machine unstable operation detection using Dragonfly® sensors

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**Abstract:** This document details a case study of a 1936 600 MVA Shock Motor Generator equipped with hydrodynamic bearings known for their ability to handle high radial forces. The main objective is to investigate an instability encountered during an attempt to increase the rotating speed, leading to "oil whirl" and "oil whip" phenomena. The study highlights the use of Dragonfly® sensors for precise deformation measurements and compares their performance with conventional methods. The results emphasize Dragonfly® ability to monitoring hydrodynamic bearings, particularly at low frequencies.

**Key Words**
F-Lab, Hydrodynamic bearings, Strain, Oil whirl & Whip, Motor generator

*Figure 1: 600 MVA Motor generator used to generate electrical shock on electromechanical equipment under test.*

## 1 Introduction

Shock motorgenerators are very specific equipment designed to stress electromechanical devices. The Schneider F-Lab, located in Grenoble, France, is linked to a few other Schneider laboratories in the world (USA, China, Spain, India, Germany) through the One Labs network. The Grenoble site, located in Technopole and Electropole, has three of these generators for testing its own equipment with 2x600 MVA and 1x2500 MVA. The machine here studied dates back from 1936 and enables high load tests such as short-circuit establishment and breaking, high currents of short duration, internal arcs, load rejection in active, inductive, capacitive or reactive mode on various test benches.

### 1.1 Hydrodynamic fluid film bearing

The technology used here to hold the shaft line and support the strong radial forces in the motor and the generator is hydrodynamic fluid film bearing. The stationary part  and rotating surfaces are separated by a thin film of lubricant. Considering an oil-lubricated journal bearing as shown in Figure 2, the relative motion between the rotating and stationary parts creates a converging wedge that produces a distributed pressure that supports the radial load (perpendicular to the axis of rotation).

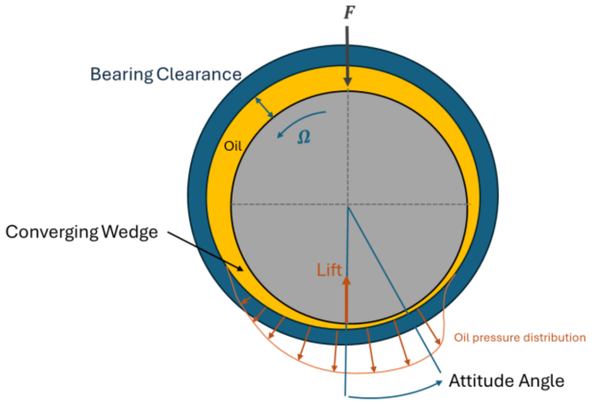



*Figure 2: Hydrodynamic bearing schematic and oil film pressure distribution. Ω is the rotation speed.*

The oil film is designed to support radial loads, dissipate heat from the film shear and provide damping to the system. Their design and construction seem relatively simple, but the theory and operation of these bearings can be quite complex. In fact, to improve its dynamic and static performances, the bearing shape is designed with careful consideration of the circumferential speed, static and dynamic load or oil ISO grade among other parameters. It is possible to find various types of technologies with one or more converging wedges known as cylindrical, lemon shape, offset, multi-wedges or tilting pads. The advantage of these designs is a more accurate alignment of the support shell to the rotating shaft and an increase in shaft stability.

## 1.2 Oil Whirl & Whip

In hydrodynamic bearings, the shaft line is maintained during its rotation by the lift of a thin film of oil as pictured in Figure 3. The position occupied (attitude angle and eccentricity) by the center of the shaft line in its bearing depends on factors like load, viscosity and rotating speed. The Figure 3 variables are:

$\boldsymbol{\Omega}$ is the rotation speed,
$\boldsymbol{F_T = jD\lambda\Omega r}$ is the tangential stiffness force,
$\boldsymbol{F_s = -K_B r}$ is the rotor-bearing system stiffness force
$\boldsymbol{F_D = -D\dot{r}}$ is the bearing viscous damping force,

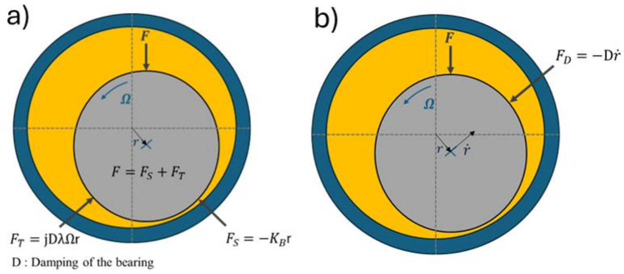



*Figure 3: Forces acting on a normal shaft line. Stiffness related forces in a), viscous force in b).*

A small rotating perturbation like a whirl imbalance is applied to the shaft line at the pulsation $\boldsymbol{\omega}$ pictured in Figure 4 with the following variables:

$\boldsymbol{m}$ is the imbalance mass.
$\boldsymbol{r_u}$ is the imbalance eccentricity,
$\boldsymbol{F_P = mr_u\omega^2 e^{j(wt+\delta)}}$ is the issued excitation force,

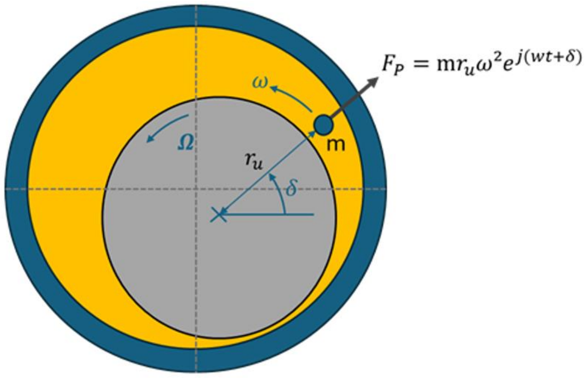



*Figure 4: Definition of a whirl imbalance excitation at ω*

**Oil whirl** is known to be the most common cause of instability in hydrodynamic fluid film bearings. Typically, the oil film itself flows around the shaft at an average speed slightly less than 50 percent (0.42 to 0.48) of the journal surface speed and, as a result, is seen in the waterfall spectra as a small rise proportional to the machine rotating speed.

**Oil whip**, a large instability of the oil film, may be observed if the resonant frequency of the shaft corresponds to the oil whirl frequency. The rotating speed at which this phenomenon occurs becomes the main critical speed of the machine. To observe oil whirl and whip, the machine must be running at more than twice the first critical speed of the main rotor shaft, which is the case here.

The nonlinear motion due to the fluid-induced instabilities can be very harmful to the system, especially the oil whip phenomenon, which confirms the need to predict the threshold of unstable motion:

We can apply the first principles associated to bearing equilibrium and we obtain, with $M$ being the rotor mass:

$$F_S + F_T + F_D + F_P = M\ddot{r}$$

If we replace radial stiffness, and damping, viscous effort and excitation described in Figure 3-a-b and Figure 4 with their expressions:

$$-K_B r + jD\lambda\Omega r - D\dot{r} + mr_u\omega^2 e^{j(wt+\delta)} = M\ddot{r}$$

$$M\ddot{r} + D\dot{r} + r(K_B + jD\lambda\Omega) = mr_u\omega^2 e^{j(wt+\delta)}$$

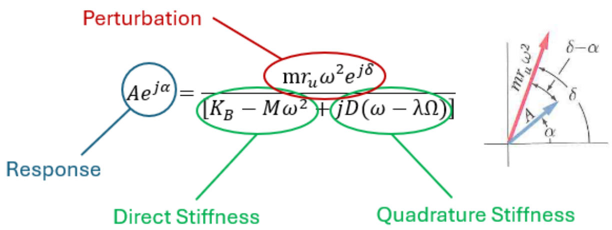
And use a response solution of the type:

$$r = Ae^{j(wt+\alpha)}$$

We finally come to the expression of the response of the shaft line:

$$Ae^{jwt}e^{j\alpha} = \frac{mr_u\omega^2 e^{jwt}e^{j\delta}}{[K_B - M\omega^2 + jD(\omega - \lambda\Omega)]}$$

In this expression, $Ae^{jwt}$ is the rotating term, $e^{j\alpha}$ and $e^{j\delta}$ are respectively the phases associated with the response and the perturbation respectively. The final shaft line response in a fluid film bearing is :




With $\boldsymbol{Ae^{j\alpha}}$ the magnitude response of the rotor shaft.

Due to the supporting hydrodynamic forces, the inertia of the rotating system and of the oil fluid damping, the shaft moves around the equilibrium position, describing an orbit. The fluid-induced instability of this rotating system depends on the relationship between these parameters (radial and tangential forces, inertia of the system and viscous damping of the oil). If both the direct (exerted radially) and the quadratic stiffness (exerted tangentially) tends to zero at the same time (i.e. when $\boldsymbol{\omega = l\Omega = \sqrt{(K/M)}}$, it can lead to an infinite response of the shaft, blocked only by the radial edge of the bearing, which increases the direct stiffness ($\boldsymbol{K}$), explaining the very excentric and erratic rotor motion in the bearing.

### 1.3 Challenges of increasing the rotational speed

The Motor generator described in Figure 1 and Figure 5 has been used to meet specific industry needs up to 16kV at 50Hz (3000 rpm), 10Hz and 16⅔ Hz (Railways) frequencies. Now, Schneider wants to address new markets at higher frequencies and needs to increase the rotor speed up to 60Hz (3600 rpm). In this project, all the necessary studies were carried out to ensure that the shaft line, coupling and new motor could handle the additional load increase and centrifugal forces associated with this upgrade.

## 2 Machine arrangement

A recently rebuilt 1700 kW induction motor is responsible for starting the power generator to full speed to provide the kinetic energy needed for the test requirements. The fluid film bearings used here to support the shaft line are lubricated by a gravity-based piping system to provide a constant pressure oil flow.

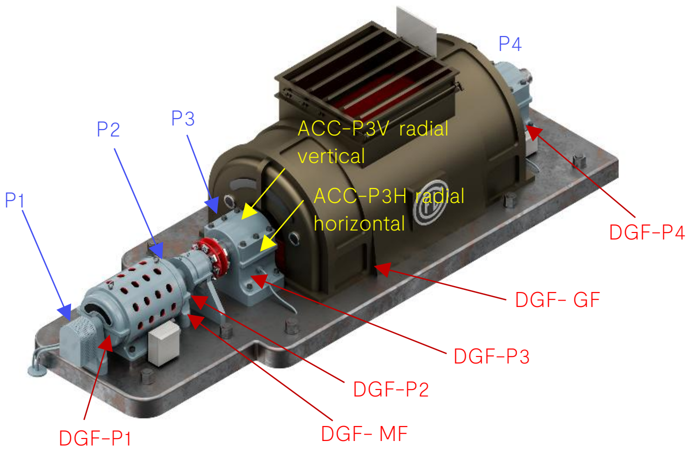



*Figure 5: Schneider 600 MVA Motorgenerator*

The lubrication system feeds three hydrodynamic journal bearings named P1, P2 and P3, and a combined journal and thrust bearing P4. Figure 5 is describing bearing names and instrumentation.

### 2.1 Dragonfly® instrumentation

Dragonfly® sensors are dynamic strain gauges made from an extremely thin crystalline piezoceramic. The sensing element is less than 10 µm thick, giving it the flexibility and ductility of a 2D material. The flexibility of the entire sensor greatly simplifies its integration on objects, and its crystalline nature results in high durability and signal quality. The ability to sense with great accuracy and sensitivity low amplitude deformation at low frequency makes it a suitable candidate for measuring film fluid bearing efforts. The IEPE version installed here has been developed to adapt the range to the signal level and to the ADC (analog to digital converter), resulting in a measurement range from 10nm/m to 3000µm/m.

Six IEPE Dragonfly® sensors(1.08mV/µm/m) and two accelerometers (IMI-100mV/g and Rockwell-500mV/g) are instrumented as described in Figure 5 and below :

**P1 :** DGF-P1 in radial horizontal direction
**P2 :** DGF-P2 in radial horizontal direction
**P3 :** DGF-P3 on anchorage vertical direction
ACC-P3V in vertical direction (100mV/g)

ACC-P3H in horizontal direction (500mV/g)
**P4:** DGF-P4 on anchorage vertical direction.

The sensors are bonded with bicomponent HBM X60 after careful paint removal and degreasing.

The generator bearing instrumentation is done close to the anchorage where we expect to have the greatest value of strain as pictured on Figure 6.

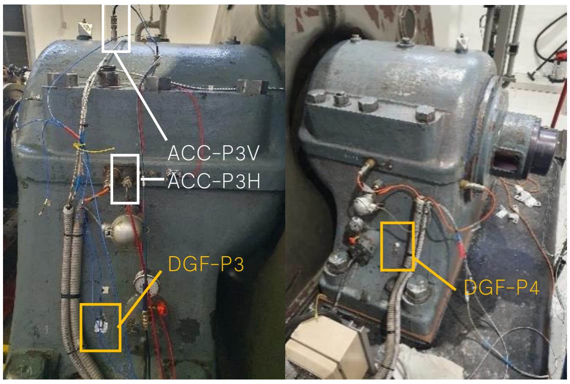



*Figure 6: P3 and P4 instrumentation*

The Dragonfly® and accelerometers are recorded using an 8 channel Dewe43 with IEPE conditioning at a sampling rate of 5kHz.

# 3 Measurements

The machine has already performed a pre-qualification run with a first run-up tentative at 3600 rpm a few days before the current test and remained at this speed for a few hours without any noticeable adverse behavior, so the tests were started with confidence.

## 3.1 Stationary measurements

FFT processing is performed at steady-state speeds from 200 to 1200 rpm. The spectra exhibit a classic response consisting of a 1X fundamental and harmonics.

The deformation values at 1X in (µm/m rms) measured on P3 and P4 are similar in magnitude. The background noise seen in the spectra Figure 7 is 1 nm/m giving 40dB of dynamic for behavior analysis (with the full range of 3000µm/m, it would represent 130dB in total).

| RPM | 200 | 300 | 400 | 500 | 600 | 700 | 800 | 1200 |
|---|---|---|---|---|---|---|---|---|
| P3 | 0.08 | 0.08 | 0.12 | 0.09 | 0.12 | 0.15 | 0.21 | 0.41 |
| P4 | 0.18 | 0.13 | 0.09 | 0.09 | 0.11 | 0.12 | 0.13 | 0.23 |

*Table 1 : 1X Bearing deformation on P3 and P4 (µm/mrms)*

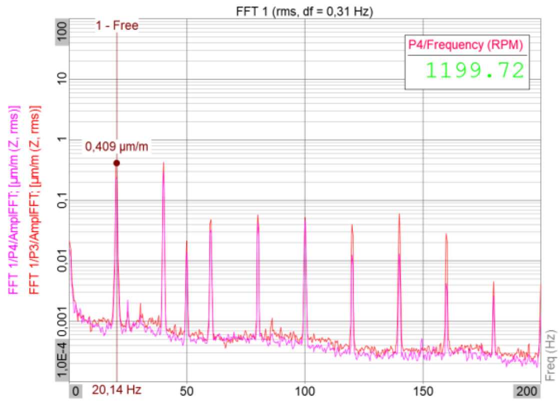



*Figure 7: stationary spectra at 1200 rpm exhibiting a classic machine response*

On Figure 8, the acceleration magnitude measured at 200 RPM on P3H is lower than 0.01 mg at 3,36Hz, the Dragonfly® sensor shows a dynamic of 37dB on the rise at the same frequency with 10 clear harmonics while the classic 100mV/g accelerometer does not even see the 1X frequency as shown below.

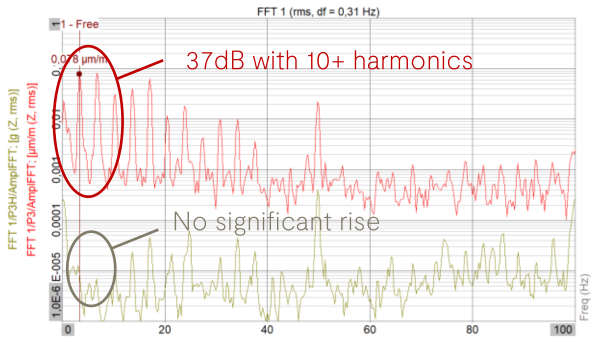



*Figure 8: low frequency response, comparison between accelerometer and deformation sensitivity*

To compare metrological quality and background noise for both sensors in the low frequency domain, a comparison between the Dragonfly® and an accelerometer (double integrated with a high-pass filter at 0.5 Hz) is presented in Figure 9. Because forces and displacements are corelated by the Hook law, it is suitable to double-integrate accelerometer and directly compare absolute displacement and micro-deformation..

At 3400 rpm, we start to see an oil whirl signature on the bearing that follows the rotating speed (0.49 times 1X) but this energy remains disorganized around 25 Hz. The horizontal absolute displacement coming from ACC-P3H (green) compared to the Dragonfly® sensor (red) shows that the piezoelectric strain sensor is much more accurate in following the whirl phenomena in the low frequency range as the bottom spectrum is lower and undeformed by the double integration, DGF-P3 bandwidth however is wider with [0.01-100kHz].

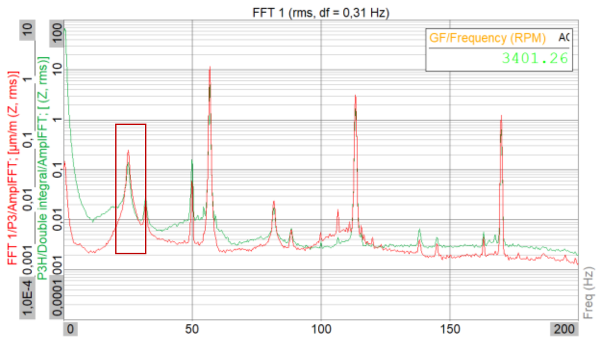



*Figure 9: 3400 rpm whirl signature on the bearing anchoring and double integrated accelerometer in horizontal direction.*

## 3.2 First ramp up

It is decided to ramp up the machine to a nominal speed of 3600 rpm where it will operate for 60Hz tests. When the speed of rotation reached 3441 rpm, the vibration amplitude at the oil whirl frequency (0.44 times 1X) has increased and even exceeded the 1X magnitude to reach 52 µm/m peak to peak with a clear excitation of the civil work. The machine was immediately shut down and an oil whip was diagnosed by examining the waterfall spectrum. Figure 10 shows the temporal signal of sensors DGF-P3 (red), DGF-P4 (purple).

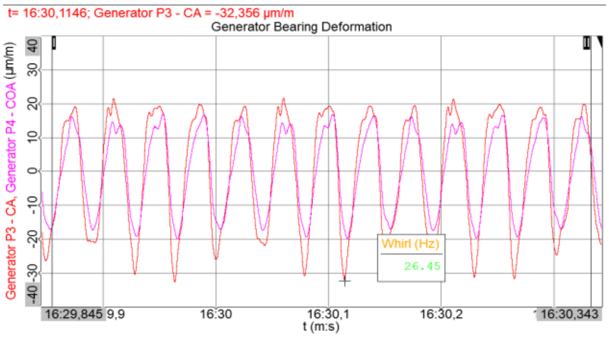



*Figure 10: deformation waveform signature on the bearing P3 and P4 (red and purple respectively) in µm/m*

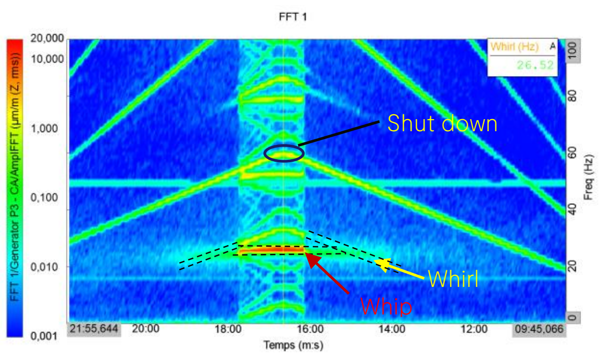



*Figure 11: Spectrogram of the Dragonfly sensor (DGF P3) showing whirl (at 14:00) and hard whip (at 16:00).*

The heat map in Figure 11 clearly shows the whirl and whip phenomena where a mode shape is excited at 25,66Hz and maintains the same frequency as the speed increases (from 3440rpm to 3600 rpm) to the setpoint and decreases after the shutdown is triggered.

In the case of the Schneider Motor generator, we can consider that the bearing is dimensioned for very high transient loads and, as a consequence, is considered as unloaded in normal operation making it work at low eccentricity and high instability condition. A few other factors are also pushing in this direction:

- Oversized bearing (designed for shocks testing)
- Large clearance (800µm)
- Bending mode shape at 27Hz (FEM)
- Designed for 3000 rpm application in 1936
- Well done alignment with small radial load
- Well balanced rotor

# 4 Standards for diagnostic and improvements proposal

The standards used today for hydrodynamic bearings are based on absolute vibration and shaft displacements:

ISO 10816 or 20816: These standards define the acceptable vibration levels for rotating machines depending on their size, their rotating speed and their type of mounting. It classifies machines into different zones according to their vibration levels. The expert reports use this standard to evaluate overall vibration velocity levels (OVL) measured on bearings.

ISO 7919: This standard specifies methods for measuring and evaluating shaft displacements on rotating machines. It provides acceptable operating limits for shaft motion based on machine rotating speed. The experts use this standard to analyze peak-to-peak vibration displacements (Spp and Smax) measured on hydrodynamic bearings.

Two types of sensors are used to perform vibration measurements:

- Accelerometers: These sensors measure the acceleration of vibrations. They are used to determine overall velocity levels (OVL) and absolute displacements levels (using double integration), and to construct vibration frequency spectra.

- Displacement (or proximity) sensors: These sensors measure the relative displacement between the shaft and bearing. They are used to determine peak-to-peak (Spp) displacements and to plot shaft orbits and shaft centerlines.

Vibration standards limitations: However, there are some challenges that are difficult to overcome with current instrumentation.

**Shaft displacements** are dependent on bearing clearance and are not necessarily a picture of unbalance or degradation, but rather a picture of bore loss history or pad tuning.

**Displacement probe targets** sometimes exhibit mechanical and/or magnetic runout, rendering them useless for metrology applications.

**Displacement probes are expensive** and often have uncharacterized resonances in support.

**Accelerometers are doubly integrated** to obtain absolute displacement. Their sensitivity is bad in the low frequency range (they are chosen so as not to overload in the high frequency range). Absolute displacement magnitude is thus not accurate at low rotation speeds.

**Orbits** are sometimes difficult to interpret, and if degrading radial misalignment reduces Spp it will increase accelerometer vibration transfer[8],[9], which can indicate a false positive imbalance.

**Bearing temperature measurements** are often in reality oil temperature measurement and are of little help in evaluating the bearing stress condition in the oil film itself.

### 4.1 Improving diagnostic with deformation

For these reasons, it is valuable to use a deformation sensor with high sensitivity in the low frequency range, where both displacement probes and accelerometers are not the best candidates to obtain accurate monitoring of the machine bearing condition over a long period of time. A strain sensor with a large dynamic range such as Dragonfly® can provide much more valuable information as an image of the operating radial and axial load for any clearance. Moreover, strain is directly related to constraints and efforts. These are the critical metrics that evaluate the risk of damage or cracking and determine the design and therefore the cost of new projects.

Other direct force measurement using classic Strain Gauges is impossible due to the high stiffness of the structures found in power generation, mining or steel industries, especially in old designs that have a higher safety factor than new equipment. The measured values are often in the background noise for conventional strain sensors (a few µm/m to a few tens of µm/m) and this is why Dragonfly® is the most adapted solution with nm/m resolution.

## 5 Conclusion

A 600MVA Motor-Generator on film fluid bearings was instrumented with 6 Dragonfly® dynamic strain sensors and two piezoelectric accelerometers to monitor the return to service after four month of overhaul. A few stationary tests have been done at low to medium rotating frequency showing a better sensitivity for the strain measurement than for the accelerometers in the low amplitudes at low frequency. Dragonfly® were better to extract the magnitude at the rotating frequencies and harmonics under 250 rpm while accelerometers failed to detect 1X and first harmonics (figure 8). It was demonstrated that Dragonfly® sensors are capable of diagnosing whirl and whip, and could therefore be used for early warning or machine shutdown, using a picture of stress instead of a picture of shaft or bearing displacements.

**Prospective**

Dragonfly® sensors can extract sub-µm/m deformations, which are rich typological information to diagnose the bearing health condition (imbalance, misalignment, overload) especially at low frequency level where accelerometers failed to detect the machine signature at its fundamental and first harmonics. These high-resolution strain sensors are the perfect match candidates for hydrodynamic bearings and especially those used in hydro turbines where the rotating frequencies can be as low as 70 rpm. Strain can be converted to Newtons or tons with proper calibration using a hydraulic jack release or impact hammer, depending on the machine arrangement and bearing size.

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