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DETECTION OF STRUCTURAL DAMAGE IN VIBROACOUSTIC ANALYSIS

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Abstract

Supports are one of the most popular structural elements in engineering. They have a wide range of applications, including in pressure gauge connectors, fixtures for photovoltaic and solar panels, and traffic signs. Supports are also used in highly complex engineering projects such as airplane wings or rotor blades. Monitoring methods for detecting and predicting the condition of support structures have become an important area of research. Structural damage to machines and machine parts can be prevented through early detection of fatigue cracks with the use of non-destructive methods. The paper proposes a method for detecting fatigue cracks along the cross-sectional area of a specimen based on selected parameters of the vibration signal. The diagnostic signal for analyses of specimen cross-sectional area was vibration acceleration, which was described with the use of the following parameters: changes in amplitude and waveform (FFT), RMS amplitude, changes in the amplitude of a vibrating sample, and changes in the phase angle of a vibrating sample. In the test stand, cross-sectional damage was caused by forces of inertia acting on the specimen. The results of the study indicate that all of the analysed parameters can be used to detect the loss of structural continuity (mechanical and fatigue cracks) in an object. An analysis of changes in the amplitude of a vibrating sample was the fastest and most comprehensive source of information.

Keywords: *damage to the cross-section of a sample, diagnostic signal for identifying damage to the cross-section of a sample, inertia*

1. Introduction

Supports are one of the most popular structural elements in engineering. They have a wide range of applications, including in pressure gauge connectors, fixtures for photovoltaic and solar panels, and traffic signs. Supports are also used in highly complex engineering projects such as airplane wings or rotor blades.

Monitoring methods for detecting and predicting the condition of support structures have become an important area of research. Structural damage to machines and machine parts can be prevented through early detection of fatigue cracks with the use of non-destructive methods. Conventional methods such as penetrant tests, magnetic inspections and ultrasonic tests have their limitations. Some of them are expensive and produce ambiguous results. Alternative methods, which identify vibration waveforms and parameters, may be a rapid, effective and convenient diagnostic tool for detecting fatigue cracks in machine parts and structural systems.

There are two main types of crack detection methods: linear methods, which investigate vibrations in an object and measure changes in modal parameters relative to initial values, and non-linear methods, which are frequency response analyses [1, 3, 10, 13-15]. In the first approach, a crack is always presumed to be open, and it is modelled as local elasticity [8]. The size of the crack and its location are analysed and described based on changes in modal parameters such as natural frequency [14], damping coefficient [15] or rigidity [7, 10]. This approach has two major limitations. Firstly, changes in free vibration frequency are significant only for large cracks, and secondly, an intentional shift of free vibration frequency cannot be fully ascribed to the cracking process and can also be caused by other factors, such as corrosive wear and relaxation [2].

It is generally believed that vibration theory is correlated with modal parameters of the system: free vibration frequency, damping and waveform. This physical system consists of the physical properties of a structural object (mass, rigidity and damping). The modelled parameters are homogeneous systems that can also be described by differential equations of motion in a physical model relative to its mass, damping, rigidity, acceleration, velocity and displacement. For this reason, all changes in modal parameters are directly proportional to changes in the physical properties of a modelled object that result from damage.

The problem of identifying structural damage based on changes in vibration has been widely addressed in the literature [4-6, 8, 12, 16, 17]. In this approach, vibration parameters of objects are described, and structural damage is defined as a function of changes in an object's structural parameters, such as rigidity and mass. The presence of structural damage influences the vibration response and the dynamic parameters of a given structural object. Dynamic parameters include natural frequencies as well as damping mode shapes and ratios. Those parameters are used to identify damage to a structural object. Early damage detection supports timely repair and extends the life of the system.

A system has to inspect and service on a short-term, mid-term and long-term basis to guarantee that it is operated safely and reliably. Rigidity is one of the main dynamic properties, which can induce changes in shape and frequency reduction mode, and it can increase the damping coefficient.

The aim of this study was to describe the fatigue cracking process and to identify diagnostic signals and their parameters, which can be used to determine the beginning and end of structural disintegration of a specimen. The proposed method can be used to determine optimal diagnostic signals and their parameters in the process of describing structural damage to elements subjected to inertia loading.

2. Analysed object

The analysed object was a pressure gauge connector for low-pressure and high-pressure liquid pipelines for controlling the parameters of the transported medium via a manometer or a pressure sensor. The medium transported by the pipeline generates vibrations, which have a negative influence on the analysed connector. As a result, the connector is susceptible to fatigue cracking above the point where the connector pipe is joined to the main pipeline.

The aim of this study was to determine diagnostic parameters, which are useful for identifying the destructive influence of inertia on the pressure gauge connector. After a preliminary analysis, a notched connector was used in the experiment. A notched model shortens the experimental period by accumulating stress at a specific point along the cross-section of the specimen.

The experimental model of a pressure gauge connector was built with the use of a connector pipe, ϕ12x1 mm, with a notch cut to the diameter of ϕ10.5 mm at the height of 12 mm above the main pipeline, ϕ38x1.5 mm. The main pipeline and the connector pipe were joined with a fillet weld. The experimental sample was made of 1.4301 stainless steel.

In the test stand, the sample was set into oscillatory motion with frequency (f) and constant displacement amplitude. The motion was monitored by two piezoelectric sensors and a crankshaft sensor. Vibrations were recorded by a data logger.

3. Research models

Due to the complex nature of the processes taking place in the examined object, the analytical model and the experimental model were synergistically combined to facilitate a multiparametric analysis of the studied phenomena. This approach supported signal acquisition and the identification of signal parameters for qualitative and quantitative analyses of the observed processes. The aim of the adopted approach was to improve the durability of the modelled pressure gauge connector.

Laboratory analyses were carried out in the test stand presented in Fig. 1. The stand for inertia loading tests comprised a generator of reciprocating motion (crank mechanism) connected to a clamp holding the analysed specimen. The system's dynamic parameters were identified with two ICP-100 piezoelectric accelerometers and a crankshaft sensor controlled by a variable frequency drive. Data were registered, visualized and processed by the KSD-400 analyser with the NI 6343 data acquisition device supported by the LabVIEW environment.

Accelerometers were mounted on the trolley of the reciprocating motion generator and on the manometer placed at one end of the test sample.

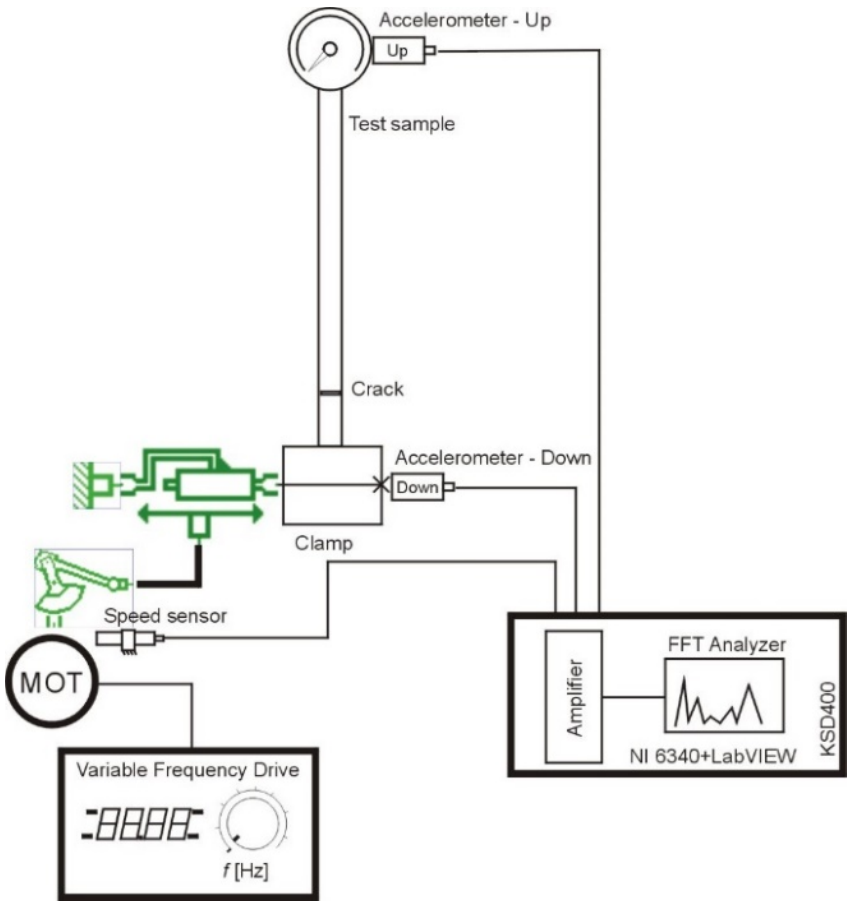

Fig. 1. Test stand for analysing pressure gauge connectors

4. Results and Discussion

The simulations performed in the AMESim environment and the experiments were conducted in conjunction. The developed mathematical model was used to perform calculations where the coefficients and values of experimental variables were input into equations. The results of tests and simulations were subjected to multiple synergistic analyses to determine time and frequency values, which are used to identify diagnostic signals and their parameters. This approach significantly reduced the time and cost of preliminary analyses.

The pressure gauge connector was tested at both low (5-30 Hz) and high (90 Hz) frequency [12]. The simulated parameters were defined during real-life measurements in a medium-pressure pumping station of a combined heat and power plant.

4.1. Experimental results

Free vibration frequency f_0, dynamic elasticity KD, static elasticity K_S and damping coefficient C (static and dynamic) for different lengths L of the connector pipe were determined in preliminary analyses. Selected dynamic parameters of the connector are presented in Fig. 2.

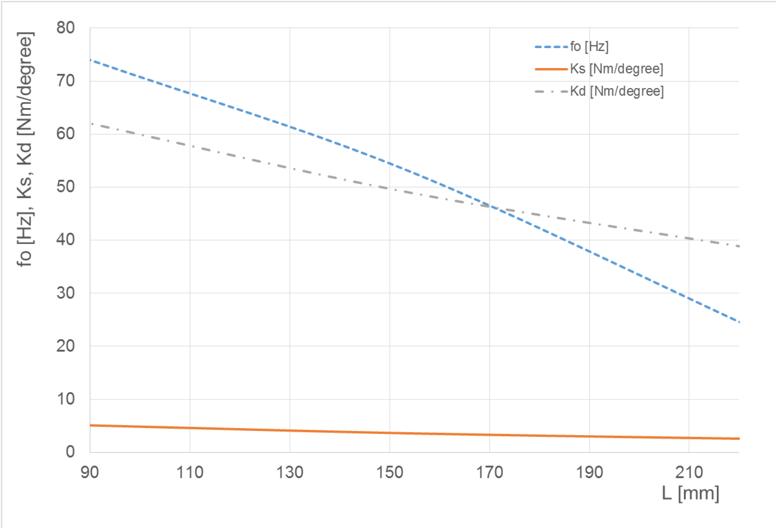

Fig. 2. Changes in free vibration frequency f_0, dynamic elasticity K_D and static elasticity K_S of the analysed connector as a function of pipe length L

For the needs of the study, the experiments were conducted at the frequency of 30 Hz, which was equal to the resonant frequency of the analysed pressure gauge connector. The applied frequency range guaranteed that the connector would be damaged in approximately 4,000 s.

The simulations conducted in the AMESim environment were experimentally validated, and the acceleration of vibrations in the free end of the connector was selected as the diagnostic signal for describing fatigue cracks on the cross-section of the sample. The following parameters of the diagnostic signal were identified:

- acceleration amplitude and vibration waveforms in the frequency domain (FFT analysis),
- RMS amplitude of vibration acceleration,
- changes in vibration acceleration at one end of the specimen,
- changes in phase angle relative to the rotational frequency of the crankshaft and the amplitude of vibration acceleration of the free end of the pressure gauge connector.

The presented results were obtained for the following test parameters: vibration frequency – 30 Hz, amplitude of vibrations in the clamp holding the sample – A=1 mm, weight of the manometer with the accelerometer – 125 g.

4.1.1. Amplitude of vibration acceleration in FFT analysis

The amplitude and frequency characteristics of the test sample's response to vibration frequency of 30 Hz, registered by the top accelerometer mounted at the manometer (in the "Up" position), are presented in Fig. 3. A frequency analysis was carried out to describe the fatigue cracking process on the circular cross-section of the sample.

At free vibration frequency (f_0), the initial amplitude of vibrations of the pressure gauge connector is constant (area A in Fig. 3), but higher harmonic frequencies ($3f_0$=90, $4f_0$=120 Hz, etc.) with constant amplitude values are also visible.

When a fatigue crack appeared on the cross-section in the notch (area B in Fig. 3), the amplitude of acceleration gradually decreased at the set frequency, and vibrations, which had not been previously identified in the vibration spectrum, were observed in the second harmonic frequency (area C in Fig. 3).

When fatigue cracking was initiated across the notch, the highest amplitude of vibration acceleration was noted for harmonic frequency. In a continuously vibrating sample, the amplitude of displacement decreased relative to its initial condition. Successive stages of sample destruction were not monitored.

Simulation results revealed that distinctive amplitudes in second harmonic frequency (area C in Fig. 3) resulted from changes in the damping coefficient of the cross-section. The dynamic (modal) parameters of the analysed connector were most influenced by the damping coefficient and free vibration frequency. The stiffness coefficient was the least significant determinant of the analysed system's behaviour.

A decrease in the amplitude of acceleration in harmonic frequency resulted from an increase in the stiffness coefficient of the cross-section of the sample. Additional energy is required for the fatigue crack to propagate along the cross-section. The energy is supplied to the structure as the system's elasticity (the stiffness coefficient describes the system, not the material) stored in the deformed cross-section of the sample. When the fatigue crack propagates inside the material, stress disappears and elastic energy is released.

Our results indicate that changes in the state of a system are first reflected by changes in damping coefficient C, followed by changes in stiffness coefficient K.

The fatigue cracking process was similar in all analysed samples regardless of the applied frequency, and the only difference was the time of crack initiation on the surface or cross-section of the sample.

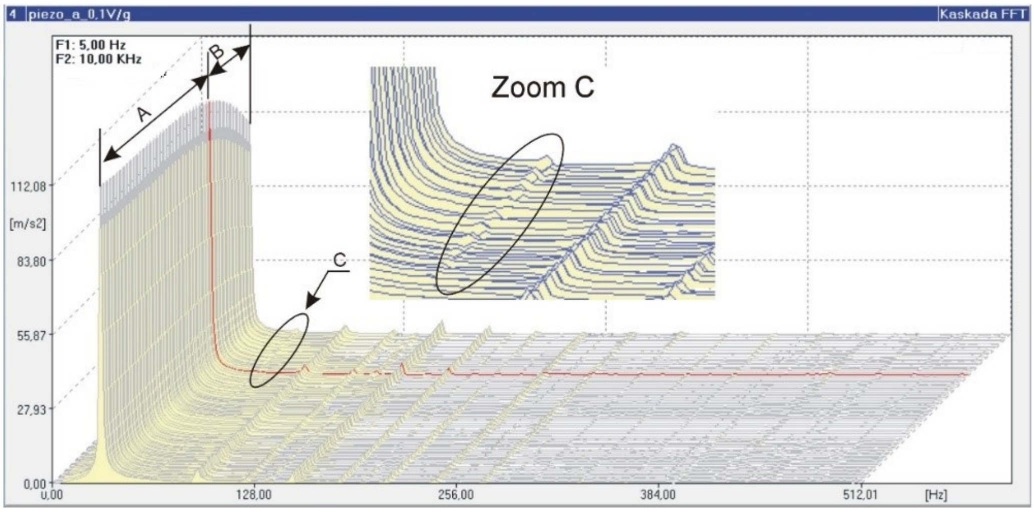

Fig. 3. Amplitude and frequency characteristics of a pressure gauge connector. Data were registered by the top accelerometer in the "Up" position. The red line denotes the initiation of fatigue cracking on the notch of the analysed sample. The weight of the pressure gauge and the accelerometer was 125 g

4.1.2. Root mean square amplitude of acceleration

The following vibration parameter was RMS acceleration measured by the top accelerometer ("Up"). Changes in RMS amplitude during the test are presented in Fig. 4 for the measured parameters and the test sample indicated in Fig. 3. The period of measurement was described by the number of cycles completed by the sample in reciprocating motion in the test stand.

An analysis of changes in RMS acceleration at one end of the sample indicates that at the beginning of the experiment, the amplitude of acceleration was constant without major fluctuations. A moderate decrease in RMS acceleration was noted after around 4,000 cycles, and a significant drop in this parameter was observed after around 6,500 cycles. An analysis of amplitude and frequency characteristics and changes in RMS amplitude indicates that fatigue cracking in the sample notch takes place after around 6,500 cycles.

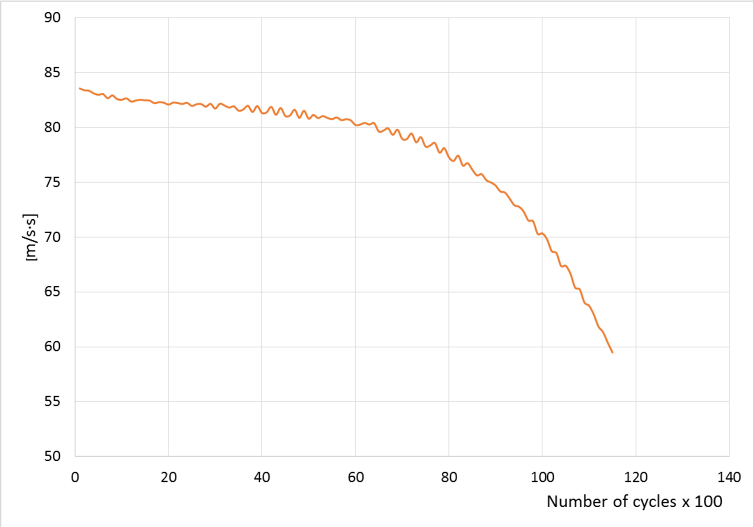

Fig. 4. Changes in RMS amplitude of acceleration of a pressure gauge connector

4.1.3. Identification of the parameters of individual deformations

Changes in acceleration amplitude of the free end of the pressure gauge connector were analysed with the use of the top accelerometer ("Up") for the same parameters that were used in the above examples. The source of vibrations was monitored, and the results were analysed in view of the acceleration amplitude of the variable frequency drive measured by the bottom accelerometer ("Down"). Changes in the acceleration amplitude of the vibration generator in selected measurement cycles are presented in Fig. 5.

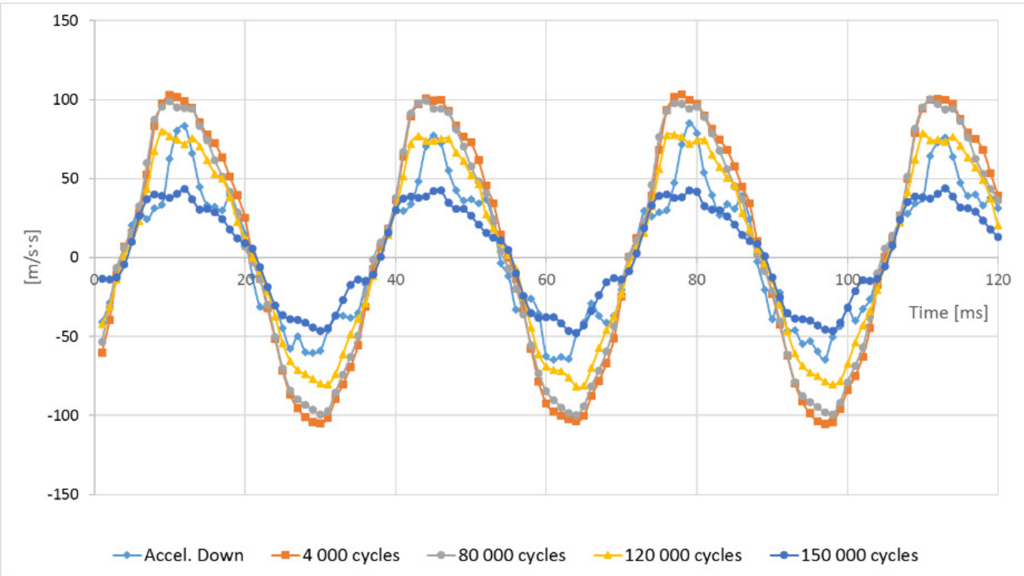

Fig. 5. Changes in acceleration of the free end of the pressure gauge connector for different numbers of cycles

An analysis of changes in acceleration of the free end of the pressure gauge connector indicate that waveform and acceleration amplitude change with the number of measurement cycles. Structural changes in the notch cross-section decreased acceleration amplitude in extreme positions of the sample and flattened the amplitude.

4.1.4. Phase angle

The last signal parameter investigated in the experiment was phase angle, which was determined based on the rotational frequency of the variable frequency drive. Changes in angle phase, which were measured by accelerometers mounted on the drive trolley ("Down") and on the pressure gauge ("Up"), are presented in Fig. 6.

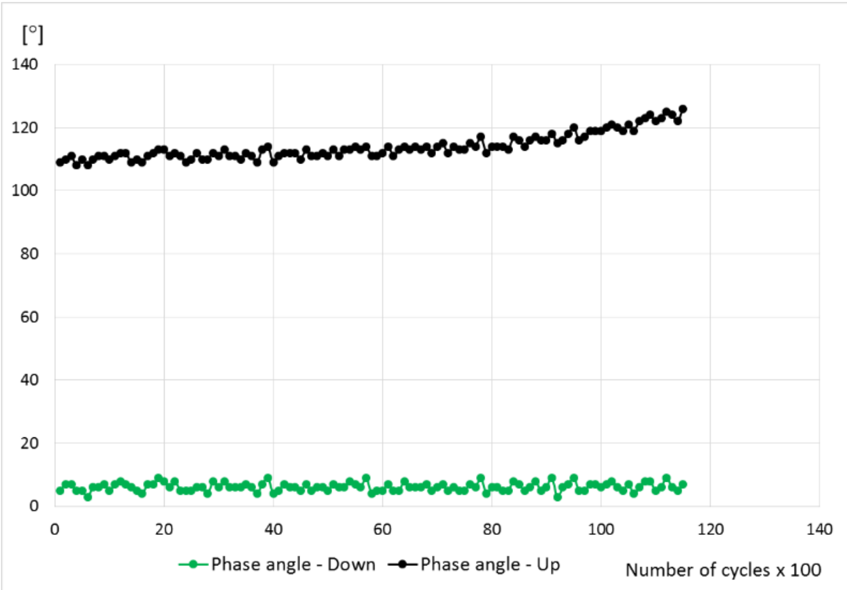

Fig. 6. Changes in phase angle measured by accelerometers in different stages of the experiment

The data presented in Fig. 6 indicate that the phase angle measured by both accelerometers was relatively constant at the beginning of the experiment. A gradual increase in phase angle was noted in the accelerometer positioned on the pressure gauge after around 6,000 cycles. Fatigue cracking in the analysed sample was initiated after 6,000-6,200 cycles.

It should be noted that the initiation of fatigue cracking cannot be determined based on the analysed signal parameters in penetrant tests. Structural microcracks on the surface of the sample are very small; therefore, penetrants are unable to soak into the structure and are not revealed when the developer is applied.

The initiation of fatigue cracking on the cross-section of the analysed sample was validated only during visual inspection under a scanning electron microscope.

5. Conclusions

The following conclusions can be formulated based on the results of the study:

- all of the analysed vibration parameters can be used to monitor and determine the structural integrity of a pressure gauge connector,
- an analysis of vibration amplitude at the free end of a pressure gauge connector was the fastest and most comprehensive source of information,
- changes in phase angle were the least critical indicator in the process of diagnosing a pressure gauge connector. The value of this parameter changed considerably during the experiment, therefore, the resulting trend was difficult to monitor,
- additional energy is required for the fatigue crack to propagate along the cross-section. The energy is supplied to the structure as the system's elasticity (the elasticity describes the system, not the material) stored in the deformed cross-section of the sample. When the fatigue crack propagates inside the material, stress disappears and elastic energy is released,

- the dynamic parameters of the analysed pressure gauge connector were most influenced by the damping coefficient and free vibration frequency, whereas the stiffness coefficient was the least significant determinant of the analysed system's behaviour,
- the initiation of fatigue cracking cannot be determined based on the analysed signal parameters in penetrant tests,
- the time until sample destruction is determined by the mechanical load applied to the cross-section of the sample. The energy required to damage a sample is described by the area under the stress curve,
- the damage to the cross-section of the analysed sample was not affected by the shape or position of the notch, and differences were observed only in vibration amplitude,
- the results of simulations revealed that vibration frequencies higher than 40 Hz did not exert a damaging effect on the analysed pressure gauge connector.

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