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Dynamic Optimization Design of the Base Plate Structure of a Track Inspection Robot based on Modal Analysis

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Abstract

The detection accuracy of track-based inspection robots is directly related to the smoothness of equipment operation on the track. Harmful vibrations caused by defects in the dynamic characteristics of the equipment structure are the fundamental reason for blurred monitoring images and distorted data. Addressing the issue of insufficient rigidity in the load-bearing base plate of a certain type of track-based inspection robot, this paper conducts analysis and parameter optimization based on the resonance risk induced by dense low-frequency modes. The parameterized finite element model, after simulation analysis using the Constrained Modal Analysis tool, reveals that the original structure's 3rd-order natural frequency within the narrow frequency band from 90 Hz to 102.60 Hz highly overlaps and is densely distributed with the drive system's second harmonic excitation frequency band from 90 Hz to 110 Hz. The same frequency can cause equipment resonance and pose a hazard to reliability. Modal analysis of the optimized structure shows significant improvement in the dynamic performance of the equipment structure, with the fundamental frequency increasing from 99.60 Hz to 186.58 Hz (an increase of 87.3%). The original dense mode group is effectively separated, the distribution of frequencies at various orders tends to be reasonable, and the natural frequencies are all migrated to the safe frequency band, resolving the resonance risk from the source. Although the optimized model weighs 0.9 kg more (an increase of 36.3%), the resulting improvement in dynamic performance is more significant. This study verifies the effectiveness of the structural dynamic design optimization method based on modal analysis, providing a reference for early risk prediction and design iteration of similar equipment.

Keywords

Modal Analysis; Structural Optimization; Inspection Robot; Natural Frequency; Resonance; Finite Element Simulation.

1. Introduction

In today's era of rapid technological advancement, industrial intelligent equipment is continuously evolving. As a novel type of industrial intelligent equipment, track-type inspection robots are now widely employed in specific settings such as substations and large factories [1-2], replacing manual labor in high-risk industries and repetitive inspection tasks. The high-definition cameras mounted on these robots enable image acquisition [3] and allow for real-time monitoring of hazardous situations within factories and identification of machinery malfunctions. Therefore, to ensure the clarity of images captured by the high-definition cameras, the stability and precision of the inspection robots'

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movement on the track play a decisive role in the reliability and smoothness of the inspection system. In practical applications, issues such as motor start-stop, impact at track joints, and eccentricity of traveling wheels [4] can cause minor vibrations during the inspection robots' operation. If the natural frequency of the robot coincides with the operating excitation frequency due to improper structural design or the aforementioned reasons, it can lead to issues such as blurred images, inaccurate detection data, and mechanical fatigue [5]. To avoid such situations, necessary designs in terms of dynamic characteristics are required. However, such designs are often lacking, resulting in vibration data anomalies being discovered only during the prototype testing phase, leading to rework and modifications, which significantly increases the equipment's development cycle and cost. In existing research, dynamic characteristics studies on track-type inspection robots focus on overall vibration reduction, control algorithm optimization, and trajectory motion planning [6], while specialized analysis and optimization schemes for the modal characteristics of the equipment's load-bearing baseplate are relatively lacking. Insufficient local stiffness of the baseplate can lead to complex coupled vibrations under broadband excitation within the same frequency band, resulting in a dense low-frequency mode phenomenon, posing numerous potential hazards and thus deserving attention [7].

This article takes the load-bearing base plate of a certain type of track inspection robot as the research object, aiming to establish a set of design and optimization processes for the base plate structure of inspection robots based on modal analysis. The first step involves utilizing the modal analysis function in finite element analysis to identify the resonance risks and modal densification issues present in the original structure. The second step involves carrying out targeted structural improvements based on the vibration modes to reduce and avoid risks. The third step involves comparing the modal parameters before and after improvement to verify the rationality and effectiveness of the structural design and optimization process.

2. Modal Analysis

2.1 Modal Analysis Theory

The purpose of modal analysis is to study the vibration characteristics of a structure under dynamic loads, encompassing parameters such as natural frequencies and vibration modes [8-9]. These parameters can directly affect the stability, safety, and lifespan of the structure [10]. Therefore, modal analysis plays an extremely important role in design optimization, fault diagnosis, and performance evaluation, whether for automobiles, robots, or precision instruments [11]. The goal of modal analysis is to identify the modal parameters of a structure, providing a basis for understanding structural vibration characteristics, diagnosing vibration faults, and optimizing the design of structural dynamics. It is commonly applied in guiding modal test design, verifying the accuracy of finite element models, and evaluating the dynamic characteristics of structures [12]. When the external excitation frequency of a system is close to or equal to its natural frequency, the system will experience significant and violent vibrations, which may even lead to unpredictable behavior. This phenomenon is commonly referred to as resonance. The excitation frequency at which resonance occurs is called the resonant frequency. Resonance does not necessarily occur at a single natural frequency, but often occurs within a frequency bandwidth within a certain range of frequencies [13].

2.2 Establishment of Finite Element Model

This paper establishes a finite element model based on the track-type inspection robot as the research object, optimizes the inspection robot model, and conceals components such as the control box and camera mounted on the inspection robot chassis, the robot shell, and the auxiliary wheels. Only the inspection robot chassis and steering mechanism are retained as the main mechanical structures optimized in this paper. The inspection robot model is imported into ANSYS software for model processing. The robot material is selected as 45# steel, and the material parameters [14] are shown in Table 1.

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Table 1. material attribute

Material properties	numerical value
Density (kg/m^3)	7850
Poisson's ratio	0.3
Elastic modulus (Pa)	2×10^{11}

Add the material properties from Table 1 to ANSYS software, set the simulation mesh to a 1mm tetrahedral mesh, and the model mesh diagram is shown in Figure 1.

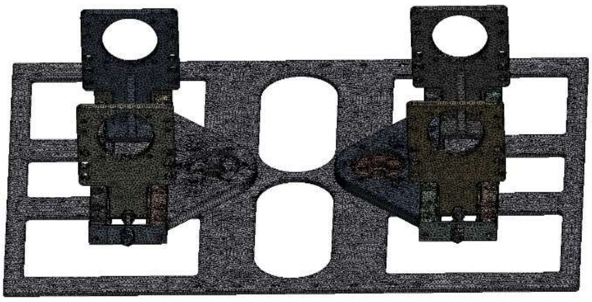


Figure 1. Model mesh diagram

After setting the boundary conditions and the contact types between various components, the modal analysis is initiated. The cylindrical surface of the steering mechanism that contacts the auxiliary wheel is fixed. During the modal analysis process, the first six modes are selected for simulation analysis [15-16]. In most engineering structures, low-order modes store and transmit the most energy during vibration. When the structure is subjected to general dynamic loads, its dynamic response is mainly composed of the superposition of the first few modes, and the contribution of high-order modes is very small and can usually be neglected. Furthermore, the first six modes can significantly reduce the computational scale, memory requirements, and storage space. Therefore, the sixth-order mode is selected for simulation analysis in the modal analysis.

2.3 Modal Result Extraction and Risk Diagnosis

After the simulation is completed, record the fixed frequencies of all orders of the model, and the data is presented in Table 2.

Table 2. Modal results of the original model

model order	Natural frequency (Hz)
1	99.60
2	101.01
3	102.60
4	113.01
5	152.55
6	193.40

The natural frequency data of each order of mode obtained through finite element simulation is presented in Table 2. The data from Table 2 is plotted as a bar graph, and the dangerous excitation frequency band range is also plotted together. The image is shown in Figure 2. The red band represents the dangerous excitation frequency band of the model.

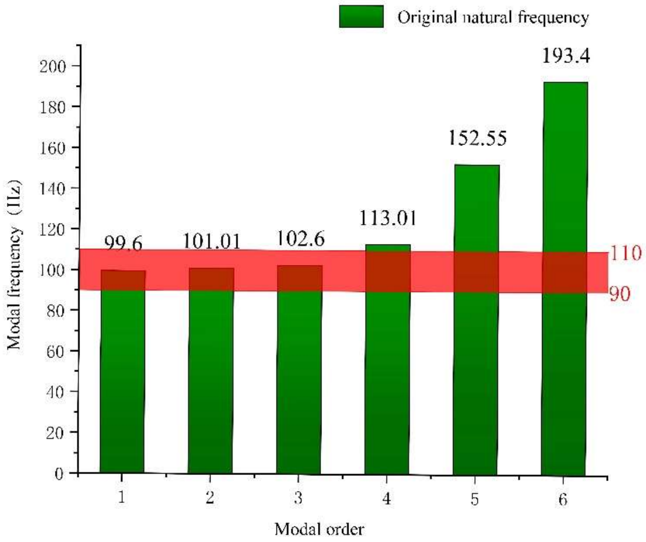


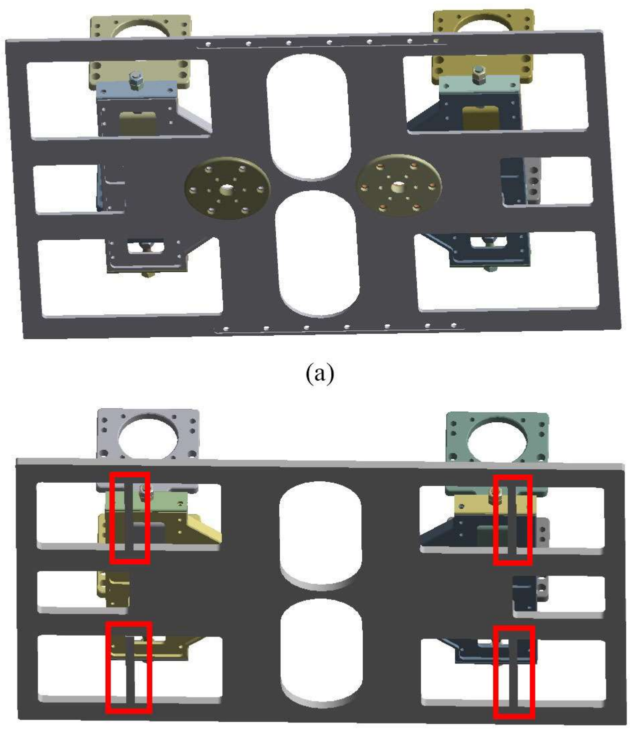
Figure 2. Model natural frequency and resonance band

Figure 2 shows the distribution diagram of the natural frequencies of the model and the image of the resonance band position. From the figure, it can be observed that the natural frequencies of the first three modes of the original structure are 99.60Hz, 101.01Hz, and 102.60Hz, respectively. The third-order modes are densely distributed within a very narrow bandwidth. From a dynamic perspective, this phenomenon indicates that within this frequency band, there are simultaneously vibration deformation modes with similar shapes but different directions shared by the first three modes [17]. When external excitation falls within this range, it will lead to complex coupled vibrations in the entire model. Such complex coupled vibrations will seriously affect the detection accuracy of the track-type inspection robot, resulting in poor image acquisition and impacting the use and control of the entire system. It can also be seen from the analysis that the first three modes are completely located in the core area of the set dangerous frequency band, indicating that the model will be simultaneously excited by the second harmonic excitation of the drive motor during operation, causing severe structural resonance and coupled deformation. This will lead to complex deformations in the base plate, resulting in loss of reliability and affecting the detection accuracy and stability of the onboard camera. In severe cases, it may even deteriorate the reliability of the robot [18].

3. Structural Improvement Design

Through the modal analysis of the original model mentioned above, it can be concluded that the natural frequency distribution of the original model is dense and there is a resonance risk with the harmonic excitation of the drive motor. Therefore, the original model is optimized based on this. The main purpose of the optimization is to increase the fundamental frequency of the model's natural frequency, avoid the dangerous zone of harmonic excitation of the drive motor, separate the dense natural frequencies of the first three modes, and improve the dynamic characteristics [19]; on this basis, it is necessary to ensure that the increase in mass is controlled within a reasonable range.

To enhance the natural fundamental frequency of the model, the thickness of the base plate was increased from 2.5mm to 5mm, thereby comprehensively improving the bending stiffness of the model. Due to the excessively large area of the hollowed-out portion of the base plate, transverse strip-shaped reinforcement ribs were added to the extensively hollowed-out region to restrain deformation of the model. The comparison between the optimized and unoptimized models is shown in Figure 3.



(a)

(b)

(a) Original model (b) Optimized model

Figure 3. Comparison chart before and after model optimization

The locations of model modifications are marked in Figure 3, including strip-shaped reinforcing ribs and a thickened base plate. The model mass increased from 2.48kg to 3.38kg. The improved model was re-imported into ANSYS software for modal analysis, using the same grid division strategy, material properties, and boundary conditions as the initial model. The changes in natural frequencies of each order of the optimized model were compared with those of the original model.

4. Presentation of Results and Verification

4.1 Modal Analysis Results after Optimization

The results obtained from the modal analysis of the optimized inspection robot base plate were compared with the modal results of the original structure. The comparison chart of the model's natural frequencies is shown in Figure 4. The histogram can visually reflect the distribution relationship between the 6th-order modal natural frequencies and the dangerous excitation frequency band of the base plate before and after optimization.

Table 3 presents a comparative table of the natural frequencies obtained from modal analysis of the model before and after optimization. The table reveals that the optimization of the model has led to an increase in the natural frequencies of all orders. Specifically, the increase in the thickness of the base plate has elevated the fundamental frequency from 99.60Hz to 186.58Hz, representing an increase of 87.3%. The first three modes of the model, which were located in the dangerous excitation band before optimization, have been successfully shifted to the safe frequency band through optimization, ensuring sufficient safety margin.

Table 3. Comparison of natural frequencies of the model before and after optimization

Modal order	Frequency before optimization (Hz)	Optimized frequency (Hz)	Increase value (Hz)	amplification
1	99.60	186.58	+86.98	87.33%
2	101.01	196.64	+95.63	94.67%
3	102.60	201.56	+98.96	96.45%
4	113.01	219.66	+106.65	94.37%
5	152.55	277.27	+124.72	81.76%
6	193.40	373.82	+180.42	93.29%

As shown in the bar graph comparing natural frequencies in Figure 4, the hazardous excitation frequency band marked with red shading completely overlaps with the first three natural frequencies of the structure before optimization, indicating a serious resonance risk that would be directly triggered by the original structural frequency and the second harmonic excitation of the drive motor. The optimized data reveals that all frequency bars have shifted from the hazardous excitation frequency band to the safe frequency band, and there is a sufficient safety margin of over 76 Hz between the lowest frequency and the upper limit of the hazardous frequency band. This margin fully meets the safety requirements of conventional design, indicating that even in the event of motor speed fluctuations or unobserved high-order harmonic excitation in practical applications, the structural frequency will not fall into the resonance zone, and motor excitation will not trigger harmful vibrations. This risk is fundamentally eliminated. The optimized structure meets basic vibration isolation requirements and significantly improves the structural dynamic stability. The reliable base plate structure lays a solid mechanical foundation for the robot to stably perform high-precision visual inspection tasks.

Moreover, the significant increase and redistribution of natural frequencies not only circumvented the known major excitation risks, but also significantly enhanced the structure's adaptability to broadband random excitation.

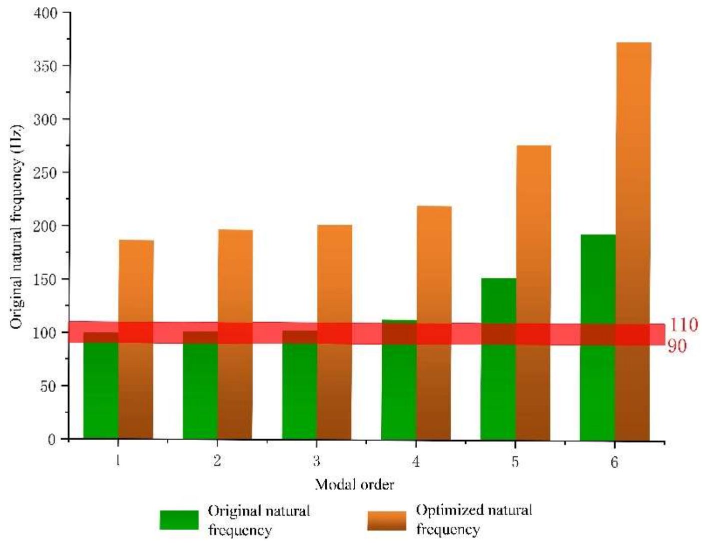
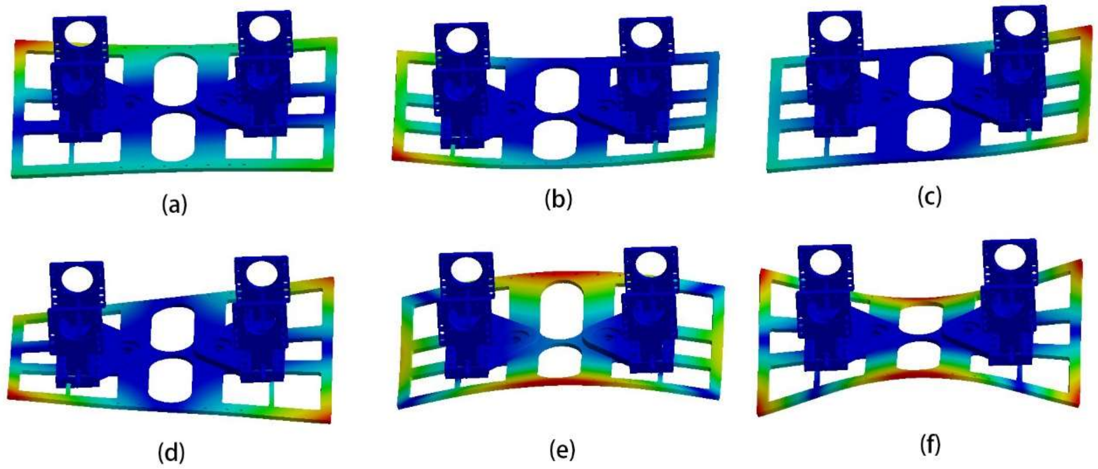


Figure 4. Comparison of natural frequencies before and after model optimization

The significant achievements of optimization are reflected in the notable increase in frequency values, and the improvement in modal distribution is also very evident. After structural optimization, the highly dense modes are effectively separated, the intervals between various modes are significantly increased, and the dynamic response characteristics become clearer and smoother. This indicates that the optimized structure helps reduce the occurrence frequency of multi-mode coupled vibrations, effectively improving the overall motion stability. Combined with experimental data, it can also be shown that before optimization, when the excitation falls within a narrow frequency range where multiple easily excited vibration modes gather, it is prone to induce complex vibrations where multiple modes are coupled, seriously affecting the dynamic response. After optimization, the minimum frequency interval of the first four modes has expanded to over 10 Hz, effectively separating the dynamic response intervals of various modes, dispersing the vibration energy distribution, and making the contribution of each mode to the overall dynamic response more independent and clear. The improvement in distribution characteristics not only reduces the risk of resonance caused by frequency pulling but also provides a clearer frequency band design reference and expected target for unobserved possible active or passive vibration control implementations. The effect can be seen in the deformation contour plots of each mode of the optimized model in Figure 5.



(a) Deformation contour plot of the 1st mode (b) Deformation contour plot of the 2nd mode (c) Deformation contour plot of the 3rd mode (d) Deformation contour plot of 4th-order mode（e) Deformation contour plot of 5th-order mode (f) Deformation contour plot of 6th-order mode

Figure 5. Deformation cloud diagrams of each order of modal of the optimized model

Based on the above review, the optimization strategy of increasing the thickness of the load-bearing base plate and adding reinforcing ribs to the base plate has significant effects in achieving the core design goals of enhancing stiffness, separating modes, and avoiding resonance.

4.2 Comprehensive Evaluation of Optimization Effect

Analyzing the natural frequency data presented in Table 3, it is evident that all natural frequencies have seen a significant increase after optimization. Specifically, the top four modal frequencies, which are most critical for performance, have all increased by over 94%. Notably, the fundamental frequency has risen from 99.60 Hz to 186.58 Hz, representing an increase of 87.3%. These data unequivocally demonstrate that the optimization strategy, which involves increasing the thickness of the base plate to 5 mm, significantly enhances the overall bending stiffness of the mechanism, serving as the primary factor behind the notable increase in frequency [20].

From the relative positional relationship between the natural frequencies before and after optimization and the hazardous excitation frequency band of 90-110 Hz shown in Figure 4, it can be observed that before optimization, the first three closely spaced modes (99.60 Hz, 101.01 Hz, 102.60 Hz) were completely within the hazardous frequency band, highly coinciding with the main excitation frequency of 100 Hz, posing a high risk of resonance. After optimization, the natural frequencies of all orders were successfully shifted to the safe frequency band, and there was a sufficient safety margin of over 76 Hz between the lowest frequency (fundamental frequency of 186.58 Hz) and the upper limit of the hazardous band (110 Hz). This fully demonstrates that the potential resonance risk caused by closely spaced modes has been completely eliminated at the design level.

Before optimization, the first three modes of the structure were densely distributed within a very narrow bandwidth of 3Hz, which easily led to complex coupled vibrations. After optimization, the frequencies of the first three modes were distributed dispersedly within the region, with the minimum interval increased to over 5Hz. The mode denseness phenomenon was effectively alleviated, and the dynamic characteristics of the structure became clearer, which is beneficial for vibration control.

Improving structural stiffness inevitably comes at the cost of increased mass. A comparison of the structural mass before and after optimization reveals that the mass of the optimized model has increased from 2.48 kg to 3.38 kg, an increase of 0.9 kg, or 36.3%. However, the 87.3% increase in the fundamental frequency of the structural mode indicates that a significant gain in dynamic performance has been achieved at a relatively small cost in mass, indirectly demonstrating the economic efficiency of this optimization strategy for engineering applications. Meanwhile, in terms of effectively suppressing bending deformation in specific directions, the addition of transverse strip-shaped reinforcement ribs further optimizes the stiffness-mass distribution.

By comparing the contour plots of modal shapes before and after optimization shown in Figure 5, it is concluded that the low-order modal shapes are primarily influenced by the overall bending degree of the base plate, but the maximum deformation displacement of the base plate at the same order has been significantly reduced. The local features that gradually emerge in the high-order modal shapes indicate that the dynamic weak links of the structure have shifted from the simple overall bending of the base plate to local connections or components. This is a normal phenomenon due to the enhancement of the overall structural rigidity [21].

4.3 Discussion on the Effectiveness of the Optimization Scheme

The scheme analysis has verified the effectiveness of the optimization scheme for structural improvement based on modal shape diagnosis. The composite strategy of increasing the overall thickness of the load-bearing base plate and adding reinforcement ribs to local large-area holes is highly efficient and precise in addressing the modal densification issue caused by insufficient bending stiffness in the original design. Simulation results show that this optimization scheme fully achieves the three core objectives of increasing the fundamental frequency, separating dense modes, and avoiding resonance risks, thus completing the optimization design.

This method comprehensively considers the strict requirements for the stability and detection accuracy of precision instruments during the operation of the track-type inspection robot. The overall natural frequency of the model is effectively improved by increasing the structural thickness and adding reinforcing ribs. The model resonance problem caused by the second-order resonance of the motor is completely eliminated, and the structural rigidity and bending resistance are also significantly optimized. It is judged that this scheme can provide a reference for the development of precision equipment that transfers various modes to avoid resonance zones.

5. Conclusion

This article focuses on the issues of unstable operation and data loss of the high-definition camera mounted on the track-based inspection robot, which may arise due to reasons such as motor start-stop, impact from track joints, and eccentricity of the driving wheel during the robot's operation. By establishing a finite element model for modal analysis, it is revealed that the natural frequencies of the original structural modes are highly concentrated and pose a resonance risk with the second-order resonant frequency of the driving motor, posing risks to the smooth operation and structural reliability of the equipment. Based on this issue, this article adopts optimization strategies such as thickening the base plate and adding transverse reinforcement ribs. After structural optimization, the originally densely distributed modal groups were successfully separated due to the increase in the structural fundamental frequency from 99.60Hz to 186.58Hz. The natural frequencies of the first six modes were all shifted to the safe frequency band, and the distribution of frequencies across all modes became more reasonable. The resonance risk was theoretically eliminated, and the overall dynamic characteristics of the structure were significantly improved. The engineering value of the optimization strategy was successfully verified.

The structural optimization method demonstrated in this paper provides an additional approach for the design of track-type inspection robots. The summarized optimization process can intuitively enhance the structural rigidity and natural frequency of the model, offering theoretical support for improving the reliability and precision of similar equipment.

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