A new algorithm of dangerous points accurately identification on mechanical structure under dynamic loads

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Abstract. Identification of dangerous moment of mechanical structure is a key process for dynamic response optimization. The dangerous moment of identification method for solution space of dynamic response based spectral element was proposed according to the insufficient of traditional method, which was difficult to accurately determine structure dangerous moment. The solution space of structure dynamic response was obtained by using the modal superposition method. The matrices of time points and their corresponding solution space values of dynamic stress were constructed. The Lagrange interpolation method was applied in the solution space domain to getting dynamic response function of high precision. Finally, the most dangerous moment of structure under dynamic loads was found by executing the global optimizer, though constructing a mathematical model to identify dangerous moment. The theoretical research results were applied to engine piston with dynamic loads to verify, and results showed that dynamic stress and dynamic displacement was not simultaneously reach extreme moment of the dynamic load, due to the nonlinear relationship between stress-strain and strain-displacement of structural. The existence of a time gap between them leads to dynamic optimization design is inaccurate. Dynamic optimization design should also consider two dangerous moments in order to avoid structure inaccurate design. This paper is the foundation of dynamic response optimization of piston.

1. Introduction

Structural dynamic optimization was difficult due to time-related constraint processing, and structural dynamic analysis itself was more complicated. In addition, dynamic optimization iterative calculation was carried out. The calculation process was cumbersome and time-consuming, and even made optimization calculation difficult to converge. The static load method [1] introduces an equivalent load to deform the structure under the static load, and the system response field formed by the stress was the same as the system response field under the dynamic load at a certain moment. The equivalent load was introduced by the equivalent principle of the system response result. Therefore, the problem of structural dynamic optimization with complex calculation and poor convergence was transformed into a static optimization problem with mature research and wide application. The proposed method provided an effective way to calculate the dynamic optimization design of complex and poorly convergent structures [2], which had become one of the main research directions of current structural optimization techniques.

However, one of the keys to the application of the method was that when the dynamic load became equivalent to static conversion? The mechanical structure was most critical at the moment of dynamic loading, in other words, at which time the structure was at the most dangerous situation, which was, the identification problem of structural dangerous moments. Therefore, it was necessary to study how to...
identify the identification of the most dangerous moments of the structure under dynamic loading. Accurate identification of key time points required constructing a relationship matrix with structural dynamic response solution space and time domain. The key to solve this matrix was to obtain an accurate interpolation function of structural dynamic response with time. In the traditional method, one of the commonly used methods was to use the empirical approach to determine the maximum value of the load as the key time point. This method was so subjective what the result often went wrong. The other method was to use the general finite element. The software adopted the uniform interpolation method, but as the number of interpolations increased, the error of the interpolation result was extremely large and invalid.

Theoretical studies had found that interpolation was performed on fixed points. If these points correspond to the zero point of the orthogonal polynomial, the obtained function got the highest interpolation accuracy [3-4]. To this end, this paper will explore an accurate identification method for structural dangerous moments under the dynamic load with high recognition accuracy and short calculation time, and strive to apply the identification method to the mechanical structure for verification.

As one of the key components of the internal combustion engine, the piston was directly related to the working reliability and durability of the high-speed internal combustion engine. Therefore, the structural design optimization of the piston had been a hot spot in the industry. After the unremitting efforts of many experts and scholars, the research on the optimization design of pistons at home and abroad had made considerable progress and obtained a lot of results [5-6]. However, due to the complex structure of the piston, it was subjected to extremely high thermal load and transient mechanical load during high-speed reciprocating motion. Therefore, the traditional static optimization [7-8] design method neglected the dynamics of the piston under the coupling of the heat engine. The influence of the response characteristics made the optimization design process difficult to approach the real working state of the piston, and the accurate identification of the most dangerous moment of the piston became the key to the realization of the dynamic response optimization of the piston. At present, there were mainly the following methods for determining the dangerous moment of the piston: ①The maximum moment of dynamic load was often used as a dangerous moment in the project, which was feasible for structural static strength evaluation, but for structural optimization design, it was easy to cause structural over-design or under-design; ②Engineers often took the maximum absolute value of displacement as the dangerous condition, and believe that as long as the working condition was satisfied, the structure meets the design requirements, and the fatigue failure caused by the dynamic response of the structure was difficult to predict and control.

2. Identification theory basis

Regarding the recognition theory of structural hazard moments, the application was studied in literature [9] that adaptive search technique, least squares parabolic spline function approximation method and super peak method in identification. The key time point identification in the literature aimed to solve the dynamic optimization problem directly to satisfy the time-dependent constraint function which was not enough to consider that the objective function was generally a time-dependent function. In dynamic optimization calculation, the solution to the objective function still needed dynamic response analysis, which was not fundamental to solve the problem; the literature [10] used the dynamic displacement response to determine the dangerous moment when the differential of time was equal to zero. However, it was difficult to solve the differential itself. The specific solution was not given in detail.

As the number of interpolations increased, the approximate value obtained by the uniformly distributed finite element method might cause an error or an approximate failure.

Under dynamic load, the dynamic stress at any point met the following conditions.

$$\sigma(t) = \sum_{k=1}^{m} \sigma_k(t)$$

Where $\sigma_k(t)$ — the kth-order modal stress at any point of the structure, obtained by equation (2).

$$\sigma_k(t) = DB_k q_k \Phi$$

Where $m$ —— the number of main modes intercepted;
$D$——elastic matrix;  
$B$——the first main vibration strain displacement relationship matrix;  
$Q$——the first main mode coordinate;  
$\Phi$——modal matrix.

The Lagrange interpolation function of the dynamic response result at the GLL point was

$$
P_i(\xi) = \frac{\Phi_{k+1}(\xi)}{(\xi - \xi_j)}\Phi_{k+1}(\xi_j)
$$

(3)

Where $\Phi_{k+1}(\xi)$——the dynamic stress interpolation basis function of the corresponding GLL point, the dynamic stress value at the GLL point was equal to the value obtained in this step;

$$
\Phi_{k+1}(\xi) = (\xi - \xi_1)(\xi - \xi_2)\cdots(\xi - \xi_k)(\xi - \xi_{k+1})
$$

(4)

Since the structural dynamic response function was unknown and its characteristics were extremely complex, generally nonlinear multi-peak functions. The global optimization algorithm was used to search the absolute maximum points of structural dynamic stress. The mathematical model for constructing dangerous moment identification was as follows.

$$
\max \sum_{i=1}^n \phi_{x(i,j)}(i)\phi_{x(j)}(j)\Phi_{k+1}(x)
$$

s.t.  
$$
\Phi_{k+1}(x) = (x - x_1)(x - x_2)\cdots(x - x_k)(x - x_{k+1})
$$

$$
x_k \in \left\{x \left(1 - x^2\right)^{1/2} k! \frac{d^k}{dx^k} \left(x^2 - 1\right)^k = 0 \right\}
$$

$$
i = 1, 2, \cdots, n - 1; \quad j = 1, 2, \cdots, m
$$

3. Piston dynamic response analysis

The identification of the dangerous moment of the piston under the dynamic load was first carried out to analyse the dynamic response of the piston. The piston was subjected to cyclic dynamic loads such as high-pressure gas pressure and inertial force generated by high-speed reciprocating motion under working conditions to generate mechanical stress and mechanical deformation. At the same time, due to the high temperature generated by the combustion of the high-pressure gas, the temperature of the top of the piston and even the entire piston was high, and the temperature distribution was very uneven, resulting in thermal stress and thermal deformation of the piston. Therefore, the piston needed to be subjected to thermal-mechanical coupling transient dynamics analysis. In this paper, a 12-cylinder V-type diesel engine piston was taken as the research object. By constructing a combined finite element analysis model of the piston and related components, the thermal-coupled dynamic response of the piston under thermal steady state was studied.

3.1. Thermal boundary conditions

The third type of boundary condition was used to calculate the temperature field of the piston. The boundary heat transfer between the piston and the surrounding material was divided into three parts: the heat exchange between the top of the piston and the high temperature gas, the heat exchange between the piston and the cooling water, and the heat exchange in the cold oil chamber of the piston. The heat transfer coefficient of each zone was calculated by the classical formula and combined with the piston temperature measurement test and numerical simulation to determine the thermal boundary conditions of the piston.

3.2. Mechanical load boundary conditions

The mechanical load of the piston mainly included the reciprocating inertial force and the gas force. Combined with the actual force of the piston, the gas force was applied to the top surface of the piston
according to the uniform force, and was applied to each ring groove and the surrounding shore in a certain proportion. In addition, the inertial force was applied by applying a piston acceleration load.

3.3. Constrained boundary conditions
In order to ensure the calculation accuracy and improve the calculation efficiency, the 1/4 finite element model was used for analysis, the symmetric displacement constraint was applied on the symmetrical section, and the full constraint was added at the bottom end of the connecting rod; the piston head and the piston body, the pin seat and the piston pin were added. Between the components that contacted each other, such as the connecting rod and the piston pin, a contact pair was established according to the assembly relationship to ensure the transmission of force and heat.

The temperature of the piston node calculated by the temperature field was applied as the body load to the corresponding node, and was applied together with the mechanical load boundary condition. The dynamic response analysis of the thermo-mechanical coupling in the three cycles of the piston was carried out, and the stress of the piston under the coupling of the heat engine was obtained. Displacement distribution: Figure 1 showed the displacement time history curve of the maximum deformation part of the piston - the edge of the fire shore; Figure 2 was the equivalent stress time history curve of the maximum stress concentration part of the piston-the upper edge of the pin hole.

4. Piston hazard identification
As mentioned above, due to the action of high-temperature gas combustion, the temperature gradient of the piston was large, resulting in great thermal stress and thermal deformation. Under the combined action of mechanical load and temperature load, the piston was stressed at the inner edge of the pin hole. Concentration, so the identification of the dangerous moment of the piston needed to be carried out from the dynamic stress and dynamic displacement angles respectively.

4.1. Dangerous moment identification based on dynamic stress
Using the recognition model shown in equation (5), the dynamic stress time history curve of a certain period calculated by the simulation was discretized by the spectral elements, and 122 GLL points were obtained. Through Lagrange interpolation, the DIRECT global optimization algorithm was used to find the best. The high-precision dynamic stress time history curve was shown in Figure 3. In view of the length of the article, the corresponding data table was omitted. The key time point was t=0.017143s, and the corresponding stress value was 157.240MPa. Because the piston dynamic response had certain specificity, in other words, the dynamic change of the dynamic stress was mainly concentrated in 0.015~0.02s. So, in order to improve the recognition efficiency, the interpolation time domain was shortened to 0.015~0.02s.

Figure 1. Piston displacement time history curve   Figure 2. Piston equivalent stress time history curve
4.2. Dangerous moment identification based on dynamic displacement
The dynamic displacement time history curve obtained by simulation was also discretely discretized, and 122 GLL points were obtained. Through Lagrange interpolation, the DIRECT global optimization algorithm was used to find the high-precision dynamic stress time history curve as shown in Figure. 4. The corresponding data table was omitted, and the dangerous time point was 0.017223s.

4.3. Analysis of dangerous moments

| Different identification methods | Dangerous moment |
|----------------------------------|-----------------|
| Stress based                     | 0.017154s       |
| Displacement based               | 0.017223s       |
| Maximum load time                | 0.017s          |

Figures 5 showed the comparison of the dangerous moments obtained by different identification methods after interpolation of different GLL points. It could be seen from the figures that the number of GLL points (interpolation times) had an influence on the accuracy of key point recognition, but had no effect on the results of the three methods; the dangerous moments obtained by the three methods were not at the same time, and there was a time gap between the maximum dynamic load (S1), the maximum...
stress time (S2) and the maximum displacement time (S3) of the piston. The dangerous moment (S3) obtained by displacement obviously lagged a little. It could also be seen that it was obviously unreasonable to take the maximum moment of dynamic load (S1) as a dangerous moment in the traditional method, but it should be considered according to the deformation characteristics of the piston. Two dangerous moments of maximum dynamic displacement and maximum dynamic stress were very important for the optimal design of piston dynamic response based on equivalent static load method.

5. Conclusion
The recognition results showed that the dynamic stress and dynamic displacement of the piston did not reach the extreme point at the same time. Therefore, in the dynamic optimization design of the piston, the maximum displacement and the maximum stress should be considered at the same time, so as to avoid the underlying design problem caused by the structure which included the maximum time of the dynamic load, the maximum stress moment and the maximum displacement moment as the transformation time point. The key problem of piston hazard identification in this paper was not only the problem that needed to be considered in the application of equivalent load method, but also the important content of predicting the most dangerous moment of structure under dynamic load. The key time point identification method could be applied to the dynamic response optimization of other mechanical structures, and could also be extended to the structural evaluation and prediction problems of mechanical structures and civil engineering.

Acknowledgments
The authors gratefully acknowledge the financial support by the National Natural Science Foundation of China (Grant No. 51605447), the Applied Basic Research Programs of Shanxi Province in China (Grant No. 201601D021085) and the Program for the Outstanding Innovative Teams of Higher Learning Institutions of Shanxi.

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