Vibration characteristic simulation and finite element analysis of vehicle compressor impeller based on ANSYS

Wen Cao
Department of Vehicle Application, Army Academy of Armored Forces, No.1 Huayuan Road, Changchun, Jilin Province, China
Corresponding author and e-mail: Wen Cao, 58986717@qq.com

Abstract. The vehicle compressor impeller of the turbocharger of the vehicle is often damaged by the vibration. Model analysis for the compressor impeller is a kind of common means to avoid resonance between the impeller and natural frequency. Based on the software of ANSYS, the mid and low order nature frequency of vehicle compressor impeller was simulated and calculated by the method of submodel, and Campbell chart was drawn according to the results. The resonance speed of the nature frequency of compressor impeller was found, thus reference for optimal design of the compressor impeller was provided.

1. Introduction
With the continuous improvement of people's requirements for internal combustion engines, turbocharging has been recognized as one of the main development directions of internal combustion engine technology. High-performance turbocharging has become a high technology in the field of internal combustion engines, which has aroused great attention in the world [1]. However, turbochargers often encounter impeller damage accidents during operation. According to statistics, damage caused by vibration accounts for the vast majority of impeller damage. Therefore, one of the most important indicators in the design of turbocharger’s structural reliability is the modal characteristics of the compressor impeller of the vehicle. The modal analysis of the compressor impeller of the vehicle to evaluate the stiffness and its dynamic characteristics of the compressor impeller of the vehicle becomes particularly important.

The methods to study the modal characteristics of the compressor impeller of vehicles mainly include theoretical calculations and experimental tests. However, experimental tests often have limitations such as long cycle time and high cost, which makes it difficult to be a comprehensive and multi-scheme research application. Therefore, the use of finite element method to conduct modal analysis on the compressor impeller or similar structure of vehicles has become one of the most important methods at present. However, at present, most scholars use a complete model for modal analysis and calculation, which greatly increases the calculation amount of modal analysis, and the efficiency is obviously not high. The sub-model analysis method can obviously improve the above problems and improve the calculation efficiency.

2. Basic theory of vibration
According to vibration theory [2], the free vibration equation of a multi-degree-of-freedom undamped system is:
\[ [M]\{u\} + [K]\{u\} = 0 \] (1)

Among them: \([M], [K], \{\ddot{u}\}, \{u\}\) are the mass matrix, the stiffness matrix, the node acceleration vector and the node displacement vector, where the corresponding content are respectively the mass matrix, stiffness matrix, node acceleration vector and node displacement vector of the impeller.

It is assumed that the vibration of each part of the impeller is a simple harmonic motion with the same frequency and phase, that is:

\[ \{u\} = \{\varphi\} \sin(\omega t + \phi) \] (2)

Put equation (2) into equation (1), and remove the term containing \(t\), we can get:

\[ \{k\}_i\{\varphi_i\} = \omega^2_i[M]\{\varphi_i\} \] (3)

Where \(\varphi_i\) is the \(i\)-th eigenvector and \(\omega_i\) is the \(i\)-th natural frequency. Solving this equation can find the vibration mode of the blade.

When the impeller is subjected to centrifugal force, a differential stiffness matrix \([S]\) is generated, and the stiffness matrix of the impeller becomes:

\[ [K_s] = [K] + [S] \] (4)

Therefore, the formula for calculating the dynamic frequency of the impeller is:

\[ [K_s]\{\varphi_i\} = \omega^2_i[M]\{\varphi_i\} \] (5)

3. Finite element simulation of vehicle impeller modal analysis

The modal calculation of the compressor impeller of the vehicle requires the following, according to the structural shape characteristics of the compressor impeller of the vehicle, considering the actual use conditions of the compressor impeller of the vehicle, taking the solution time and accuracy as the basic scale:

(1) The structure of the compressor impeller of the vehicle is complex, and the model should be simplified without affecting the dynamic characteristics of the structure;

(2) The finite element model grid must have sufficient density to ensure the accuracy of the calculation results and truly reflect the modal characteristics of the compressor impeller of the vehicle.

3.1. Establishment of the compressor impeller sub-model of the vehicle

Because the impeller structure of the compressor of the vehicle is cyclically symmetrical, so it is only necessary to analyze one cycle using the method of sub-model analysis. The three-dimensional design software SolidWorks is used to divide the impeller into 6 sub-models, for each of which 60 degrees is a rotation period, and then the sub-models can be assembled into a complete model by means of geometric expansion. The sub-model structure diagram and the complete model diagram are shown in Figure 1 and Figure 2.
Figure 1. Structure of sub-model.

Figure 2. Structure of whole model.

3.2. Convergence analysis of the modal calculation of the compressor impeller of the vehicle and the establishment of the finite element model

In the finite element calculation of the modal of compressor impeller of the vehicle, it is difficult and labor-intensive to select the high-order unit to realize the division of the finite element mesh, due to the complexity of the compressor impeller structure of the vehicle. Adopting the idea of non-manifold modeling with the use of tetrahedral ten-node quadratic elements can reduce manual intervention in the model. Therefore, in this study, such element discretization is used in the finite element mesh model, and the focus of the convergence analysis is concentrated on the discussion of the selection of element feature size.

With reference to the overall size of the compressor impeller structure of the vehicles, the element side lengths $h$ are selected as 3mm, 2mm, 1mm, 1.5mm and 1mm respectively, and the finite element mesh is divided. The number of nodes and elements of the finite element model are shown in Table 1.
Table 1. The numbers of elements, nodes and second-order nature frequency about different element size

| Element side length(mm) | 4     | 3     | 2     | 1.5   | 1     |
|-------------------------|-------|-------|-------|-------|-------|
| Number of elements      | 956   | 1669  | 4117  | 8403  | 26293 |
| Number of nodes         | 2004  | 3290  | 7531  | 14439 | 42721 |
| Second-order frequency(Hz) | 15700 | 15438 | 15264 | 15198 | 15171 |

The Lanczos algorithm was used to solve the finite element model of the compressor impeller of the vehicle with different levels of refinement. In order to ensure the calculation accuracy in the low and middle frequency domain, the low-order calculation results of the compressor impeller of the vehicle were the focus for convergence analysis. Table 1 only shows the second-order natural frequency value of the compressor impeller of the vehicle with a rotating speed of 100,000 r/min.

![Figure 3](image-url)  
Figure 3. Convergence law of frequency change with element size.

Figure 3 is the fitted relationship curve of the second-order natural frequency of the compressor impeller of the vehicle with the element size. It can be seen from the curve that the calculated value of the natural frequency gradually decreases as the element size decreases, and there is a tendency to converge to a certain value. It shows that the use of tetrahedral ten-node elements for finite element mode calculation has monotonic convergence, which is consistent with the theory of finite element solution accuracy. Considering the calculation scale and solution accuracy comprehensively, the use of a tetrahedral ten-node solid element with a size of 1.5 mm for modal calculation of the compressor and turbine impeller can meet the requirements of calculation accuracy. The use of a smaller element size will not bring about error reduction in calculation, but it can increase the calculation scale to an unacceptable level. The finite element model divided according to the element size of 1.5mm is shown in Figure 4.
Figure 4. Meshs of sub-model.

4. Discussion and analysis of modal calculation results for the compressor impeller of vehicle

When using a sub-model structure for modal finite element analysis of the compressor impeller of the vehicle, the period symmetry analysis option needs to be defined. In the result, the natural frequencies of each order of 0–3 knot diameters was be obtained. The natural frequency and the corresponding vibration modes have the greatest influence on the dynamic characteristics of the system, so only the first 5 order modes of the compressor impeller of the vehicle are calculated. Tables 2 and 3 list the first 5 order frequency calculation result of the compressor impeller of the vehicle at 0 pitch diameter and the first 5 order frequency calculation result of each pitch diameter when the rotation number is 110000 r/min respectively.

Table 2. The first 5 order frequency of 0 nodal diameter at different rotate speed (Hz).

| Order     | Rotating speed (r/min) |
|-----------|------------------------|
|           | 0          | 100000 | 110000 | 120000 | 130000 |
| First order | 4473.6     | 4665.8 | 4832.4 | 4999.0 | 5165.6 |
| Second order | 14968     | 15198  | 15246  | 15298  | 15354  |
| Third order   | 16688     | 16915  | 16962  | 17013  | 17068  |
| Fourth order  | 19343     | 19424  | 19447  | 19467  | 19488  |
| Fifth order   | 22979     | 23467  | 23567  | 23677  | 23795  |

It can be analyzed from the data in Table 2 that the presence of the rotating speed increases the corresponding frequency. For the same order frequency, the higher the rotating speed, the higher the corresponding frequency; at the same rotating speed, the centrifugal force has a greater impact on the lower order frequency but has less impact on the higher order frequency.
Table 3. The frequency of 0—3 nodal diameter at the rotating speed of 110000r/min (Hz).

| Order       | Pitch     |          |          |          |
|-------------|-----------|----------|----------|----------|
|             |           | 0        | 1        | 2        | 3        |
| First order |           | 4832.4   | 0.12042E-01 | 14968   | 15003    |
| Second order|           | 15246    | 0.13501E-01 | 14968   | 16596    |
| Third order |           | 16962    | 8199.4   | 16560   | 21916    |
| Fourth order|           | 19447    | 8199.4   | 16560   | 22511    |
| Fifth order |           | 23567    | 15235    | 20585   | 28187    |

It can be seen from Table 3 that repetition frequencies appear in different order modes at the same speed, and the reason is because the structure of the wheel disc and the side bars are symmetrical, and the same vibration mode and frequency with different phases may occur. It should be noted that when the impeller speed is 110000r/min, the first 2 orders frequency before 1 pitch diameter is very small, which is a rigid body mode, and the frequency can be regarded as zero, which is not considered when performing resonance analysis.

Figure 5. Campbell chart.

Figure 5 is the Campbell diagram of the impeller (Campbell diagram). Because the analysis of the high-order frequency of the impeller is not significant, so only the first 5 natural frequencies of the impeller are used to draw the Campbell diagram. It can be seen from the figure that the compressor impeller of the vehicle intersects with the second-order natural frequency at 114596r/min under the action of 8 times the excitation force within the range of working speed 110000r/min—120000r/min. Under the action of 9 times the excitation force, it intersects with the third-order natural frequency at 113261r/min. Under the action of 10 times the excitation force intersects it with the fourth-order natural frequency at 116903r/min. Under the action of 12 times the excitation force, it intersects the fifth-order natural frequency at 118445r/min. This shows that the impeller tends to resonate when operating at the above rotating speed. In order to avoid this phenomenon, the natural frequency of the impeller or the frequency of the excitation force should be adjusted.
5. Conclusions
(1) There are a total of four resonance rotating speeds within the working speed range of the impeller, and there are certain problems in the design. In order to avoid the resonance of the compressor impeller of vehicles, the structure of the impeller blade can be changed within the range of satisfying its natural frequency, and the dispersion of the blade can be reduced as much as possible to avoid the excitation frequency of the blade falling within the range of its resonance frequency.
(2) Through the modal analysis of the vehicle compressor impeller of turbocharger, it can be seen that the modal frequency of the impeller is slightly higher when considering prestress than that when we donot considering prestress. This is because the prestress generated by the centrifugal force on the impeller movement increases the stiffness of the impeller and increases the modal frequency under operating conditions, but with little effect.
(3) The sub-model method is used for modal analysis, and the low-order natural frequency solution of the overall model can be obtained more accurately by applying periodic symmetry constraints.

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