Dynamic finite element model validation of an assembled aero-engine casing

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Abstract. Structural dynamic model updating and validation of an aero-engine casing is critical to the design and development of an aircraft engine. It helps to identify the dynamic characteristics and reduce the response of the aero-engine. The modelling and parameter identification of joint are extremely difficult and important in structural dynamic analysis of the assembled aero-engine casing. In this paper, dynamic model validation technique was applied to update and validate the finite element model of an assembled aero-engine casing. First, modal test of individual casings and the assembled casing was performed by using the traditional acceleration sensors and a hammer. The modal frequencies and mode shapes were obtained by modal analysis tools. Second, the Inverse Eigen-sensitivity Method was used to correct frequency errors and MAC values of correlated mode pairs in the individual components to obtain validated models. Last, the bolt joints of two aero-engine casings were modelled by thin layer of shell elements. The material parameters or element properties of the thin-layer contact elements were updated to obtain reliable connection parameters. The results show that the errors of natural frequencies between the validated FE model of an assembled aero-engine casing and test data are within 7%, and the MAC values of main modes are above 70%, which can verify the feasibility and effectiveness of this approach.

1. Introduction
With the rapid development of computer technology, finite element models have been widely used to analyze and predict the dynamic characteristics of components and the whole machines, gradually replacing the test of the actual structure. As discretization and simplification of an actual structure, the finite element model has some uncertainties in the process of modelling such as material properties, geometry and boundary conditions etc. The difference between the simulation results and the test data of the actual prototype seriously affected the credibility of dynamics with FE model simulation, which hindered its wide application in the design process of aero-engine structures. In order to improve the accuracy of the dynamic model, the general practice is to verify and update the finite element model by using the vibration test data of the actual structure as a reference, so that the validated model has enough accuracy to reflect the dynamic characteristics of the actual structure, providing reliable guarantee for dynamic analysis and design of structure.

The concept of model validation is clearly defined in the monograph written by Ewins [1]. In practice, the validation of finite element dynamic models is mainly based on comparing experimental modal analysis results with analytical predictions. Several requirements for a validated model were defined by Link [2]: (1) The model must be capable of predicting the experimental modal data or the frequency response functions (FRFs); (2) Prediction of the modes beyond the active frequency range;
(3) Prediction of FRFs obtained from other loading conditions or boundary conditions than those used for analytical model updating. Model updating and validation of aero engine components and the whole model is carried out, and the framework of the model validation is proposed in Ref. [3, 4].

Model validation procedure contains several main steps: model verification, correlation, model updating. Model updating is the most crucial step of model validation, which aims at increasing the correlation between the dynamic characteristic of a reference model and that of the prediction from the finite element model. The sensitivity-based model updating method is widely used in engineering applications [5, 6]. Generally, the model updating requires obtaining acceptable correlation between the two data sets while maintaining physically meaningful parameter values. For this purpose, setting an objective function and selecting updating parameters are key processes of model updating. In conventional model updating procedures, an objective function is set as the weighted sum of several residuals, such as eigenvalue residuals and eigenvector residuals. Also the success of finite element model updating depends heavily on the selection of updating parameters. Mostly in sensitivity-based model updating, the high sensitive parameter is selected as the updating parameter, but not a real uncertain or error parameter. Several parameter selection methods were proposed in Ref. [7-9], but they still need to be further studied.

The prediction of the dynamic characteristic of general flange bolted joints in aero engine casing has attracted much attention due to its influence on the dynamics behaviour of the engine response. The simplest approach to model by a bolted joint is a rigid connection between the contact flanges. This leads to a stiff connection and higher resonance frequencies because of ignorance of all the joints influence. An improved joint modelling is the flexibility link element, such as spring and beam element. The most accurate joint modelling is the 3D bolted model and contact theory to demonstrate the dynamic and static characteristic of the joint structure [10]. Modelling joints as flexible elements or 3D bolted components is too computationally expensive for most practical engineering applications, and simplified methods are often necessary. Friswell proposed a layer of special interface elements having material properties that may be adjusted to improve the prediction of the joint structures [11]. Joint models indemnified by this approach are found to be valid in engineering.

In this paper, validation of the jointed aero-engine structure is studied, that is composed of a high-pressure compressor casing, a stator support structure and a combustor casing. Because of complexity of the assembled casing, a three-step strategy is proposed in validation process. First, modal tests of these three individual casings and the assembled casing were separately performed by using the traditional acceleration sensors and a hammer to obtain their modal frequencies and mode shapes. Second, model updating was carried out on each of single casings separately in order to validate the casing component models. Last, the assembled casing was updated with emphasis on updating joint parameters. In the actual jointed structure, there is an interference fit between the stator support structure and combustion chamber casing, and the three components are assembled by bolts, so that bolt joint modelling is needed to simulate the complex connection relationship. The joints were modelled using a layer of continuous shell elements that have material properties to be adjusted. The material parameters or element properties of the thin-layer contact elements were updated to obtain reliable connection parameters. Through these steps, the finite model of the assembled aero-engine casing was validated, laying the foundation for engineering applications such as structural response prediction, optimal design, and reliability analysis and so on.

2. METHODOLOGY OF MODEL VALIDATION

2.1. Concept of model validation

The purpose of dynamic model validation is to improve the finite element model of an actual structure, so that it can accurately reflect the dynamic characteristics of the structure, and later the model can be applied to structural design optimization. The structural dynamic model validation procedure can be described in Figure 1. Clearly, the model validation procedure generally includes several steps such as modal test, model verification, correlation and updating based on reference data.
2.2. Correlation of the experimental and the predicted models

Model correlation is one of the major steps of model validation. It is usually used to evaluate the extent of similarities between the predicted data from a FE model and the reference data. The reference data can be obtained from modal test of a real structure or from a virtual supermodel. In this paper, the reference data was supplied by the experimental modal test of a structure. The correlation step can be used to confirm correlated mode pairs between the predicted and reference data, laying a foundation for model updating. Many correlation analysis methods have been proposed, and some of these are described in Ref. [1]. Modal Assurance Criterion (MAC) is the most commonly used in engineering. It is defined by:

\[
MAC(A, X) = \frac{|\psi_X^T \psi_A|^2}{(\psi_X^T \psi_X)(\psi_A^T \psi_A)}
\]  

(1)

Where \(\psi_X\) is the experimentally-measured mode shapes; \(\psi_A\) is the theoretically-predicted mode shapes. Apparently, the MAC value is a real scalar from zero to unity. If the experimental and theoretical mode shapes are in fact from the same mode, the MAC value of 1 is expected. When the repeated modes exist in the axial symmetric structure, the MAC isn’t sufficient to characterize the pair modes between the test and the analytical. Rotating the test or analytical mode shapes around the symmetric axis may help to improve the correlation results.

2.3. Model updating: Inverse Eigen-sensitivity Method

Model updating is a key step in the process of model validation. According to different objectives, the traditional methods can be divided into direct and indirect methods. In the direct methods, the individual elements in the system matrices are adjusted directly from comparison between test data and initial analytical model predictions. These proposed methods are simple and direct, but there is no guarantee that the updated parameters are physically meaningful. In the indirect methods, changes are made to specific physical or elemental properties in the model in a search for an adjustment which brings measured and predicted properties closer together. Several kinds of updating methods can be found in Ref. [5].

The inverse eigen-sensitivity method overcomes the main drawback of direct methods, which can provide a correct result with physical meaning, so it is widely used in engineering. This method is based on sensitivity and the objective function is to be built first to represent the difference between the analytical solution and the test data for a physical system response. This difference is expressed by

Figure 1. Model validation process
the residual in the objective function, that is the relative change of the structural modal characteristics, and then to minimize the difference by some optimization algorithms. In the process of implementation, the sensitivity matrix is first established. At the same time, because of the nonlinear relationship between the updating parameters and modal characteristics, the model parameters are updated by the iterative method.

Assuming that the difference of modal parameters between test and finite element calculation can be described with its sensitivities, this can be expressed as follows:

\[
\Delta \omega_i^2 \approx \sum_{s=1}^L \frac{\partial \omega_i}{\partial \omega_n} a_s + \sum_{s=1}^L \frac{\partial \omega_i}{\partial \theta_s} b_s = \sum_{s=1}^L \frac{\partial \omega_i}{\partial \theta_s} \theta_s
\]

(2)

Where

\[
\Delta \Psi_r = \sum_{s=1}^L \frac{\partial |\Psi_A|}{\partial m_s} a_s + \sum_{s=1}^L \frac{\partial |\Psi_A|}{\partial k_s} b_s = \sum_{s=1}^L \frac{\partial |\Psi_A|}{\partial \theta_s} \theta_s
\]

If data are available relating to measured and predicted values of several \((m)\) modes of the structure, then \(m\) sets of equations of the type shown in equation (2) can be derived and assembled together into the single equation below:

\[
\begin{bmatrix}
\frac{\partial \omega_{A1}}{\partial m_1} & \cdots & \frac{\partial \omega_{A1}}{\partial m_L} & \frac{\partial \omega_{A1}}{\partial k_1} & \cdots & \frac{\partial \omega_{A1}}{\partial k_L} \\
\frac{\partial |\Psi_A|}{\partial m_1} & \cdots & \frac{\partial |\Psi_A|}{\partial m_L} & \frac{\partial |\Psi_A|}{\partial k_1} & \cdots & \frac{\partial |\Psi_A|}{\partial k_L} \\
\vdots & \ddots & \vdots & \vdots & \ddots & \vdots \\
\frac{\partial \omega_{Am}}{\partial m_1} & \cdots & \frac{\partial \omega_{Am}}{\partial m_L} & \frac{\partial \omega_{Am}}{\partial k_1} & \cdots & \frac{\partial \omega_{Am}}{\partial k_L} \\
\frac{\partial |\Psi_A|}{\partial m_1} & \cdots & \frac{\partial |\Psi_A|}{\partial m_L} & \frac{\partial |\Psi_A|}{\partial k_1} & \cdots & \frac{\partial |\Psi_A|}{\partial k_L}
\end{bmatrix}
\begin{bmatrix}
a_1 \\
a_L \\
\vdots \\
b_1 \\
b_L
\end{bmatrix}
= \begin{bmatrix}
\Delta \omega_i^2 \\
\Delta \omega_i^2 \\
\vdots \\
\Delta \omega_i^2 \\
\Delta \omega_i^2
\end{bmatrix}
\]

(4)

It can be abbreviated to:

\[
[A]_{m(n+1) \times 2L} \{p\}_{2L \times 1} = \{\delta\}_{m(n+1) \times 1}
\]

(5)

Where \(m\) is the number of correlated mode pairs that are available, \(n\) is the number of measured response points for each mode shape, \(2L\) is the number of model parameters that are considered to need updating. It is clear that if the equation (5) is over-determination, a solution can be found. However, the matrix \(A\) may be singular if not all the test data are independent, which can be done by computing a generalized inverse for \(A\). Then we can write:

\[
\{p\} = [A]^+ \{\delta\}
\]

(6)

in which the solution obtained for the required correction factor \(\{p\}\) is a minimum least-squares solutions.

2.4. Joint modeling using a thin layer contact element

Modelling of the bolted flange joints between two aero engine casings was performed by a thin layer of elements. The validity of the model method is restricted to elastic closed-state joints. The thin layer of joint elements in the present model has isotropic material properties, which would allow for very considerable adjustment of the joint behaviour including couple effects between normal and shear stiffness terms if this could be physically justified. The bolt connection model is suitable to simulate the linear feature, not suitable for the nonlinear characteristic.

The linear constitutive equation for thin layer elements may be written generally in the form
Then, assuming that all the thin layer element properties remain constant, there are 21 parameters for element, i.e. $C_{ij}, i=1,\ldots,6, j=1,\ldots,6$, to be identified. This assumption is reasonable if the flange is joined by many identical bolts. In order to reduce the updating parameter number, isotropic material was adopted. This approach was performed by two steps; first determine the thickness of the layer element to adjust the modal test result, then updating the material properties to get a more accurate result.

3. Case study: an aero-engine casing assembly

3.1. Modal testing of individual casings and joint casing

A modal test was performed using traditional hammer excitation and acceleration transducers. The Casing was fastened with four rubber bands at the four proportional spacing bolt holes. The vibration signals were measured by the PCB transducers, and recorded by the four channels DP data acquisition unit. The ICATS modal test and analysis tool is used to obtain the natural frequencies, damping ratios, and mode shapes, with the frequency response functions (FRFs) generated by the DP software. Because of the approximate symmetric character of the casing structures, two transducers were located with an angle to identify the repeated modes. All modes within 1000Hz were considered, especially those within 500Hz. Figure 2 shows the experimental configuration of the Combustor Casing and the test grid of the measurement points. Table 1 shows the tested natural frequencies and Figure 3 shows the experimental mode shapes of Combustor Casing.

![Figure 2](image_url)

(a) Experimental setup for modal testing of the Combustor Casing

(b) Figure 2. Experimental setup for modal testing of the Combustor Casing

Table 1. The 1st-10th natural frequencies of Combustor Casing

| Mode | 1st  | 2nd  | 3rd  | 4th  | 5th  |
|------|------|------|------|------|------|
|      | 62.9Hz | 64.0Hz | 169.3Hz | 170.1Hz | 292.5Hz |
| 6th  | 299.7Hz | 312.1Hz | 313.2Hz | 423.7Hz | 427.9Hz |
Similarly, the High Pressure Compressor Casing modal experiment was setup and natural frequency and mode shape was obtained by the same way. Figure 4 shows the experimental configuration of the High Pressure Compressor Casing and the Figure 5 shows the experimental modes of High Pressure Compressor Casing. The test results shows are listed in Table 2.

**Figure 4.** Experimental setup for modal testing of the High Pressure Compressor Casing

**Figure 5.** Measured mode shapes of High Pressure Compressor Casing

| 1st | 2nd | 3rd | 4th | 5th |
|-----|-----|-----|-----|-----|
| 96.4Hz | 98.9Hz | 167.2Hz | 178.2Hz | 263.1Hz |
| 6th | 7th | 8th | 9th | 10th |
| 279.2Hz | 450.6Hz | 470.4Hz | 487.3Hz | 509.3Hz |

Three components (High Pressure Compressor casing, stator support structure and Combustor casing) are assembled by 72 uniform distributed bolts. Every bolt has the same geometry and pretightening load. There are fifteen modes within 500Hz. Figure 6 shows the experimental configuration of the
joint casing and the measurement modes. The test results are listed in Table 3 and the mode shapes are plotted in Figure 7.

![Figure 6](image1.png)  
![Figure 7](image2.png)

**Figure 6.** Experimental setup for modal testing of the assemble casing

**Table 3.** The 1\(^{\text{st}}\)-15\(^{\text{th}}\) natural frequencies of the assemble casing

| Mode   | 1\(^{\text{st}}\)   | 2\(^{\text{nd}}\)   | 3\(^{\text{rd}}\)   | 4\(^{\text{th}}\)   | 5\(^{\text{th}}\)   |
|--------|-------------------|-------------------|-------------------|-------------------|-------------------|
|        | 121.0Hz           | 126.5Hz           | 237.4Hz           | 238.9Hz           | 266.8Hz           |
| 6\(^{\text{th}}\) | 268.3Hz           | 298.9Hz           | 307.4Hz           | 374.9Hz           | 383.2Hz           |
| 11\(^{\text{th}}\) | 424.3Hz           | 428.1Hz           | 464.9Hz           | 487.8Hz           | 500.4Hz           |

**Figure 7.** Measured mode shapes of the assemble casing

3.2. Model updating and validation of individual aero-engine casings

Before model validation of the assembled aero-engine casing, three components need to be updated and validated. The combustor casing geometry model is shown in Figure 8(a). Modal parameters including natural frequencies and mode shapes were obtained after conducting a modal test on the real structure. The simplified FE model shown in Figure 8(b) can be obtained from the geometry model after simplifying the holes, bosses, chamfers that have little effect on the dynamic characteristics of the overall structure. Modal analysis of the finite element model of the combustor casing was carried out to obtain natural frequencies and mode shapes in the frequency range of 1-500Hz. The first ten mode shapes from the FE model are plotted in Figure 9. Correlation between the measured data and
the predicted data from FE model was analysed and the results were listed in Table 4. Clearly, there are three mode pairs (modes 1&2, 7&8 and 9&10) swapped because of the axial symmetry of the casing structure. The maximum frequency difference for the 7th and 8th modes reached up to 27%. From the Figure 9, it is clear that the main vibration area of these two modes were around the stator blade part. Therefore, the updating areas of the FE model were partitioned into three parts which are plotted in Figure 8(b) by different colours. The elastic moduli of these three parts were updated by using the inverse Eigen-sensitivity method.

![Figure 8. Combustor casing: (a) Geometry model; (b) FE model](image)

![Figure 9. The 1st-10th analytical mode shapes of combustor casing](image)

| Mode | Updated frequencies/Hz | NFD/% | MAC/% |
|------|------------------------|-------|-------|
|      | Update frequencies/Hz   | Initial | Updated | Initial | Updated |
| 1    | 64.9                   | 7.2    | 3.3    | 76.4    | 76.2    |
| 2    | 64.8                   | 5.2    | 1.4    | 75.9    | 76.3    |
| 3    | 169.7                  | 12     | 0.2    | 87.7    | 88.0    |
| 4    | 169.8                  | 0.7    | -0.2   | 87.6    | 87.2    |
| 5    | 293.7                  | 2.1    | 0.4    | 74.8    | 90.1    |
| 6    | 294.0                  | -0.2   | -1.9   | 74.4    | 90.0    |
| 7    | 312.9                  | 27.1   | 0.3    | 67.7    | 73.5    |
| 8    | 312.4                  | 26.4   | -0.3   | 70.3    | 74.1    |
| 9    | 427.4                  | 2.9    | 0.9    | 59.9    | 88.3    |
| 10   | 426.9                  | 1.8    | -0.2   | 60.5    | 88.5    |

The updating process for combustor casing FE model is shown in Figure 10. In the left corner of this figure, SET1 represents test data and SET2 represents predicted data from FE model. As can be seen from the changes of frequency differences, MAC values and updating parameters for each iteration, these parameters tend to be convergent after five iterations. Correlation of the test and the analytical
data from the updated model is also listed in Table 4. The changes in the elastic modulus before and after updating are shown in Table 5. The second and third parts of the elastic modulus were reduced by 39% and 35%. It is because that these two parts including a large number of holes and bosses in the real structure and they were simplified by the FE model. These two parts are also main vibration area of the 7th & 8th modes with maximum frequency differences before updating. The MAC values after updating have a certain range of improvement and can be guaranteed to be above 73%. At the same time, the frequency differences have been greatly reduced from 27.1% to 3.3%. Especially the frequencies errors of the 7th and 8th modes were reduced from 27.1% to 0.3%.

Table 5. Changes of updating parameters

| Updating parameters | Initial/Gpa | Updated/Gpa |
|---------------------|-------------|-------------|
| E1                  | 204.9       | 197.0       |
| E2                  | 204.9       | 124.0       |
| E3                  | 204.9       | 133.0       |

The model updating and validation of the high pressure compressor casing follow the same rules above. The geometry and FE model are shown in Figure 11. The high pressure compressor casing was connected of two half casing by 34 bolts. The main modes of the high pressure compressor casing are nodal diameter vibration, the bolt joint was modelling by rigid connection. The updating areas of the high pressure compressor casing FE model were partitioned into 4 parts according to the geometric simplification which are plotted in Figure 11(b) by different colours. The first to third part of high pressure compressor casing was highly simplified, so the elastic modulus needs to be updated. The fourth part of the casing was considered separately because the bolt joint has a potential influence on the casing dynamic properties.
Figure 11. High pressure compressor casing: (a) Geometry model; (b) FE model

Figure 12. The 1st-10th analytical mode shapes of high pressure compressor casing

Table 6. Comparison for high pressure compressor casing before and after updating

| Mode | Updated frequencies/Hz | NFD/% | MAC/% |
|------|-------------------------|-------|-------|
|      | Initial | Updated | Initial | Updated |
| 1    | 96.2    | 27.8 | -0.3 | 41.5 | 69.4 |
| 2    | 101.3   | 26.0 | 2.5 | 34.0 | 63.4 |
| 3    | 171.0   | 25.0 | 2.3 | 77.2 | 78.2 |
| 4    | 179.5   | 14.3 | 0.7 | 76.1 | 78.0 |
| 5    | 270.7   | 30.8 | 2.9 | 87.5 | 87.2 |
| 6    | 288.8   | 25.5 | 3.3 | 87.2 | 86.6 |
| 7    | 448.1   | 24.1 | -0.6 | 87.1 | 86.0 |
| 8    | 465.1   | 16.2 | -1.1 | 78.8 | 81.9 |
| 9    | 484.0   | 25.8 | -0.7 | 75.6 | 82.7 |
| 10   | 508.1   | 22.3 | -0.2 | 69.9 | 85.8 |

The updating process for high pressure compressor casing is shown in Figure 13. The changes of pair modes frequency error, MAC values and updating parameters tend to be convergent after five iterations. Correlation of the test and the analytical data from the updated model is listed in Table 6. The MAC values of several modes after updating have a great improvement. At the same time, the maximum frequency difference has been greatly reduced from 30.8% to 3.3%.
Through the updating of individual casing, the updated material parameters can be used for the further modelling process of assembled casings. The individual casing is validated, the only uncertainty in the assembled casing structure is the joint parameter.

3.3. Bolt joint modeling of joint casing structure
In the actual structure, three components (high-pressure compressor casing, stator support structure and combustor casing) are assembled by bolts, so that bolt joint modelling is needed to simulate the complex connection relationship. Thin layer element modelling is used there in the connection parts of dynamic. By changing the material properties of the thin layer elements, the connection of the casing under different pre tightening forces can be simulated. The simplified FE model of the assembled aero-engine casing is shown in Figure 14, wherein the finite element models of each component have already been updated and validated in the previous process. As there is an interference fit between the stator support structure and combustion chamber casing, rigid connection is adopted. The stator support structure and the high-pressure compressor casing are connected by joints that are modelled by a layer of continuous shell elements. The rigid connection between the thin layer elements and the two structures are respectively used. The finite element model of the assembled casing is obtained by these steps.

3.4. Model updating and validation of an assembled aero-engine casing
The thickness and the material parameters of thin layer elements are uncertain, so we need to carry out parameter identification based on the modal test data. Because the thickness of the thin layer elements
will affect the stiffness characteristics of the joints more than the material properties of shell elements, it should be changed in order to find the best thin film thickness. When the thickness of the thin layer is set to 0.1, 0.5, 1, 1.5, 5, 10mm, separately the first ten modes of the assembled model were calculated respectively. The correlation analyses between these different models and the test model were respectively carried out, and the curves of the frequency difference for the first ten modes with the thickness of the thin layer were obtained, as plotted in Figure 15. It can be found that the influence of the thickness of the thin layer on each order is different. Because the 9th and 10th modes vibrate obviously of the joint region, which mean the mode frequency is high sensitivity to the thickness of the thin layer. Clearly, Figure 15 shows the changes of the frequency difference with the different thickness of the thin layer. Also the figure shows that the optimized value of the two curves for the 9th and 10th modes exists when the thickness of the thin layer is 1mm in which the other modes also have lower frequency differences. It means that thin layer thickness of 1mm can be well applied to simulate the stiffness of the assembled casing in actual modal test.

![Figure 15](image_url)

Figure 15. Frequency difference with the thickness of thin layer.

The thickness of the thin layer elements has been determined, and then it is necessary to study the influence of material parameters on the dynamic characteristics of the assembled model by model updating for the thin layer elements. The initial material parameters of the thin layer elements are set to: thickness 1mm, elastic modulus 193GPa, Poisson's ratio 0.3, density 7900Kg/m³. The updating process for assembled casing FE model is shown in Figure 16. The analytical results of the FE model and the test data before and after updating are shown in Table 8. As shown in Table 8, the elastic modulus of the thin layer element is increased by 82%, which has a great influence on the 9th and 10th modes and makes the frequency difference fall from 4% to 0.3%. However, the elastic modulus of the thin layer elements has little effect on the frequency differences and MAC values of other modes. The errors of natural frequencies between the FE model of the assembled casing and test data are within 7%, and the MAC values of main modes are above 70%, so that this model can be accepted as the validated one of the assembled casing.
Table 7. Comparison for assembled casing before and after updating

| Mode | Test frequencies/Hz | NFD/% | MAC/% |
|------|---------------------|-------|-------|
|      |                     | Initial | Updated | Initial | Updated |
| 1    | 121.0               | 6.7    | 6.8    | 78.3    | 78.4    |
| 2    | 126.5               | 4.7    | 4.9    | 78.2    | 78.3    |
| 3    | 237.4               | 0.3    | 1.6    | 87.1    | 87.2    |
| 4    | 239.0               | -1.1   | 0.2    | 87.1    | 87.1    |
| 5    | 266.9               | 2.4    | 2.8    | 75.6    | 77.1    |
| 6    | 268.3               | 2.0    | 2.4    | 74.6    | 76.1    |
| 7    | 298.9               | 0.8    | 0.9    | 89.7    | 89.7    |
| 8    | 307.4               | -1.8   | -1.7   | 90.7    | 90.6    |
| 9    | 374.9               | -3.2   | -0.3   | 71.4    | 68.2    |
| 10   | 383.2               | -4.0   | -0.5   | 71.2    | 69.3    |
| 11   | 424.3               | 0.9    | 0.9    | 87.3    | 87.3    |
| 12   | 428.1               | -0.1   | -0.1   | 88.9    | 88.9    |
| 13   | 464.9               | -1.2   | -1.0   | 71.2    | 71.0    |
| 14   | 487.8               | 6.2    | 6.3    | 43.3    | 43.7    |

Figure 16. Model updating process for assembled casing

Table 8. Changes of updating parameter

| Updating parameters                  | Initial/GPa | Updated/GPa |
|--------------------------------------|-------------|-------------|
| Elastic modulus of thin layer elements | 193.0       | 353.0       |

4. Conclusions
Dynamic model validation technique was applied to update and validate the finite element models of aero-engine components and assembled structure. Modal test of individual casings and assembled casing was performed by the traditional method to get natural frequencies and mode shapes. The simplified FE model of individual casing obtained from the geometry model after simplifying the holes, bosses, chamfers that have little effect on the dynamic characteristics of the overall structure. Finite element analysis and model updating results of individuals are quite matched with the test results. The maximum frequency error of all modes less than 3.3% within 500Hz.
Moreover, the thin layer element modelling method of bolt joint of assembled casing was demonstrated. Because of the individual casing has already been validated, the only uncertain parameter of the assemble casing is the joint parameter. First, different thickness of thin layer element were taken considered to identify the proper thickness. Then, updating the stiffness parameter, i.e. Young’s modulus, is considered to get a more accurate result. This approach can accurately simulate the dynamic characteristics of the model, and quite suitable for engineering application.

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