Modeling Rig Dynamics to Determine Dynamics of GTE Elements’ Performance under Shaker Conditions

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Abstract. During shaker tests to determine the strength properties of the test article, it can be difficult to shake the article to the required in the concentrator stresses. This may be due to the losses of the force transmitted from the shaker to the test article, which occur due to an improperly designed rig. Using some methods for rig designing, along with numerical modeling, provides the possibility to make an easy-to-excite rig with minimum loss of transmitted forces. This paper deals with the modeling of the “article – rig – shaker” system executing constrained oscillations under the influence of harmonic excitation. The model’s response dependencies on dynamic excitation caused by the oscillation frequency have been outlined. The excitation level’s influence on the dynamic parameters of the model have been analyzed.

Keywords: numerical simulation, constrained oscillations, rig, shaker, oscillation decrement, gas turbine engines (GTE).

1. Introduction.

Determining the material endurance limit on a full-scale article in laboratory conditions is a basic study that confirms the service period of a GTE-mounted component. Blade is the most common GTE test article for endurance limit studies. Electrodynamic shakers are used for testing, where the blade is fastened in the auxiliary facility (rig), which is mounted directly on the table of the shaker. In order to achieve the required amplitude of oscillation (level of dynamic stress), the tests are performed in a resonance mode. It is often the case that during the tests it is impossible to achieve the required high dynamic stress level, even though high-power shakers (force rating from 1500 kgf) are used and the tests are performed in resonance mode [1-11]. It is assumed that this is due to the negative impact of bending moments, shear force, and etc. that are transmitted from the blade to the shaker. As a result, most of the shaker power is spent on resisting drag forces and the shaker starts to overheat, which may result in failure of expensive equipment. In order to avoid this problem, special rigs [4-7] are used, natural frequencies of which are similar to those of the test article and are easy-to-excite. There are no universally applicable approaches to designing such rigs in technical literature, there is also no information on a mathematical model, that can predict the rig’s performance capabilities.

One of the approaches to increasing the resonance amplitude of oscillations and reducing the lateral displacements of the rig is using a balancing structure [12]. This structure compensates for the bending moment from the test article oscillations resulting in a Q-factor improvement of the whole system.

There are shakers that are designed in such a way as to enable the study of dynamic behavior of the test article simultaneously avoiding the mentioned above problems. An example of such a structure is the shaker described in the patent [12]. The structure consists of several circular-section beams with masses at the ends in the form of weights, the mass of which is adjusted depending on the test article (Figure 1). According to the patent description, this structure compensates for bending and torsion...
moments transmitted from the test articles through the rig. Thus, the operation of the shaker is not negatively affected, hence the Q-factor of the system improves.

![Figure 1. A schematic of the shaker for fatigue tests equipped with two center line rigid rods with weights at loose ends](image)

The main advantage of this arrangement is that any article can be tested. By adjusting the mass of the balancing weights and changing the distance from the weights to the exciter axis, it is possible to compensate for the high bending moment of the test article. In addition, the geometry of this arrangement is not complex and can be implemented in a large number of test benches.

Another approach to achieving the required oscillation amplitude level is using the rig designed under the principle of a dynamic damper. In terms of the structure the rig is designed so that its natural frequency is the same as the natural frequency of the article under examination. At the same time, the rig should be easily excitable and have a much larger weight than the test article. As a result, the resonance amplitude of the test article oscillations can be achieved at low levels of input energy from the shaker, which will allow the use of low-force shakers.

Figure 2 shows the rig designed as a dynamic damper [13]. The structure is made in the form of a clamp on a thin leg, which acts as an elastic element.

![Figure 2. Fan blade simulator plate on the shaker table with a mock-up damping device](image)

Since the experimental adjustment of the rig often requires high financial and labor costs, it is reasonable to develop a computational model of the “shaker – rig – blade” system. The model shall take into account the vibration load-applying unit (loader), because its dynamic parameters influence the excitability of the whole system [1-3].

This paper discusses two types of rigs: a rig with balanced weights, and a rig based on a dynamic damper.

Through an example of modeling the rig configuration with balanced weights, we illustrate the method of considering the shaker power loss when transmitting lateral oscillations to the shaker armature.

Through an example of modeling the rig configuration based on a dynamic damper we illustrate design features, the rig is deemed to be more advanced compared to the previous version.
2. Materials and Method

2.1 The rig configuration with balanced weights model.

During the first stage of the computational model development it makes sense to examine the balanced rig proposed for testing the blades of various dimensions [12]. The model under consideration will be generated in a simplified form, using two-dimensional formulation of the problem. The full equation of motion was used to determine the dynamic parameters of the studied article, [14-15]. The equation contains all the summands, including the damping matrix. The matrix was filled in taking into account damping in the elastic elements and damping in the material.

The computational model employs the material from the Structural steel package database as the material of the whole structure.

According to the patent assumptions and descriptions, we will introduce the balancing weights into the system to reduce the lateral oscillations of the shaker armature and to increase the amplitude of the blade simulator oscillations. The mass of the weights is selected depending on the test article. A distinctive feature of the proposed computational model is the use of structural elements within it, to take into account the armature distortion’s effect on the Q-factor of the entire system. The armature distortion is induced by the bending moment transmitted from the test article during oscillations.

The computational structure shall meet the following requirements:
- the test article’s response shall be sufficient to determine the structural performance;
- a shaker armature distortion that occurs shall be minimum.

The finite element model of the shaker structure is provided in Figure 3:

![Finite element model](image)

Figure 3. Finite element model: a) without balancing weights; b) with balancing weights

The model is generated in a 2-D formulation. For the structural elements, such as a blade simulator -1, rig-2, shaker armature (coil) -3, we use rectangular plates of different sizes, taken together, illustrated as a perfectly elastic body, suspended on the springs (elastic elements) - 4, to the non-deforming, fixed baseplate -5 with a given level of damping, where the springs damping will be responsible for the Q-factor decrease. The stiffness of the elastic elements of the hangers is taken from the shaker’s datasheet [16]. As balancing weights -6, located at the ends of the rig beams, the Mass21 finite elements are used in this FE model. This element is a point element with up to six degrees of freedom: three-axis displacements, and rotations about the same three axes. Each direction of the coordinate system can be specified with different values of masses and moments of inertia[17].

An increase in the damping coefficient in elastic elements results in the reduction of resonance amplitude of simulator oscillations, impairs the Q-factor of the system and its excitability. In the numerical model, the damping values are selected so that they comply with the worst Q-factor of the system. Figure 4 shows a graph of the blade simulator oscillation amplitude change versus the specified damping in the elastic elements of the system.
Based on the results of the computations, a damping value equal to 3 was selected, at which the lowest value of the blade simulator oscillation amplitude is observed, thus fitting the worst Q-factor.

In order to identify the required mass of weights that are attached to the arms, a harmonic analysis was carried out. A graph of dependence of the largest blade simulator oscillation amplitude, with the lowest value of lateral displacements of the shaker armature, on the mass of weights was plotted (Figure 5):

Based on the results of the computations, mass for each weight was defined as equal to 0.761, whereby the maximum oscillation amplitude of the blade simulator is observed.

2.2. The rig configuration based on the dynamic damper model

The version where the natural frequency of the rig oscillation is the same as the natural frequency of the blade oscillation is one of the commonly used versions of the structural arrangement of the experimental facility. In this case, the blade will play the role of the dynamic damper of oscillations. The force that arises in the shaker will be transmitted to the blade with minimum losses.

Let’s consider the behavior of the “shaker – rig – test article” system, where the structure simulating real articles to the maximum will act as a computational model.

The computational model of the shaker is presented in the form of a perfectly rigid body consisting of imitators of full-scale pieces of the structure, such as a rig clamp, a leg, a faceplate, a moving element of the vibration loader (shaker armature). Each structural element has mass characteristics and stiffness identical to those of full-scale pieces. The rig is designed in such a way that its natural frequency consistent with the first longitudinal mode is close to the natural frequency of the tested blade that is in turn consistent with the first flexural mode. However, in order to reduce the transmission of bending moments and shear forces from the blade to the shaker armature, the first flexural modes of the “leg-clamp” system shall be as low as possible by the oscillation frequency, than
that of the blade. In this case the principle of vibration isolation of spurious oscillations will be implemented [18]. If the clamp has relatively high mass it will be difficult to use the circular-section leg to meet the vibration isolation requirement. The obvious solution would be to use a rectangular leg with the required longitudinal stiffness and low bending stiffness.

The blade in the experiment is a dynamic damper of rig oscillations, and oscillates with high, required resonance amplitude levels. Figure 6 demonstrates the computational scheme of such a shaker and the FE mesh to determine the dynamic parameters of the rig at the stage of the rig adjustment before tests.

As a boundary condition (BC) of the computational model, the rotation of a simulated element of the shaker armature is disabled, as in reality the shaker is not designed to allow this. The other degrees of freedom take place due to the geometrical features of structural elements of the system. To define natural frequencies and modes of oscillations, the finite element (FE) mesh with the size of elements arranged in two elements by thickness and with square approximation of displacements is used. This FE mesh enables reproducing the movements of the oscillating system with desired accuracy.

3. Results

3.1. The rig configuration with balanced weights model

The harmonic analysis of the structure at the same load as the previous version has revealed an increase in the oscillation amplitude of the blade simulator by 10 times. The results of the computations are shown in Figure 7:
3.2. The rig configuration based on the dynamic damper model

As a result of the numerical calculations, the first 10 natural frequencies and modes of the computational model (Modal analysis) were determined. During experimental studies of blade flexural modes, it is necessary that the rig executes longitudinal mode oscillations. In this case, the previously described dynamic damper principle is implemented and the experimental unit operates with minimum power loss. Flexural modes of the rig should be avoided, as they result in the swinging of the entire moving structure, including the shaker armature, thus negatively affecting its performance, even if the test article is not heavy and large.

It is possible to excite the flexural mode of the test article in the most effective way, if the rig oscillates in the first longitudinal mode. In this case, the maximum oscillation amplitude of the rig and the test article is achieved, which facilitates the achievement of the required level of stress-strain state in the test article’s concentrators.

Calculated longitudinal modes of the rig under different computation conditions are shown in Figure 8.
Figure 8. First natural longitudinal mode of oscillation of the computational model under different computation conditions: a) computation without shaker armature; b) computation with shaker armature with resilient mounting of the armature and rig.

It should be noted that determining the dynamic parameters of the rig without taking into account the shaker armature results in a significant difference from the frequencies obtained experimentally, while taking it into account in the structural design of the shaker armature results in closer values of oscillation frequencies.

In order to assess the level of excitation of the test article, we will calculate the system’s response to dynamic excitation (Harmonic response). The level of dynamic excitation is selected from the technical data of the shaker being used to test the full-scale structure, for which the computational model is generated.

To determine the accurate value of oscillation amplitude, the damping coefficient of the oscillations is set. It is identified from the laboratory experiments on defining the damping decrement.

As a result, an amplitude-frequency response of the test article and rig has been obtained. The amplitude-frequency response graphs are shown in Figure 9.

The natural frequency of the rig is selected so that it is close to the natural frequency of the test article. Therefore, the principle of dynamic damper is implemented, where the test article acts as a dynamic damper. At the same time, there is a significant increase in the resonance amplitude of oscillations of the test article and minimum oscillations of the rig.

Thus, considering this model takes into account the effect of the vibration loader, it is possible to achieve near-to-experimental values of the test article displacements on exposure to the harmonic load.
4. Conclusions

The paper presents computational models of the shaker, taking into account the armature distortion when simulating the laboratory experiment.

Applying the principles of static balancing of the rig structure results in the improvement of the dynamic behavior of the experimental facility and reduction of the input energy losses.

In order to determine natural oscillation frequencies of complex structures, such as the considered design of the shaker, it is necessary to take more accurate account of the specific performance features of moving parts of the shaker. In this case, the difference in the obtained results will tend to zero.

In the future, it is planned to further refine the structural design of the shaker by introducing the elastic elements responsible for the suspension of the shaker armature. The stiffness and damping coefficient in the elastic elements will be determined experimentally.

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