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Experimental study on dynamic coupling characteristics of mid turbine frame

Zhongbin Chen[1, a],Zhun Liu[1, b]， Mingfu Liao[1, c,*]

[1]Northwestern Polytechnical University School of power and energy Xi'an,China

[a]2021262028@mail.nwpu.edu.cn，[b]liuzhun@mail.nwpu.edu.cn,[c]mfliao@nwpu.edu.cn

Abstract. Aiming at the problem of excessive vibration of a dual-rotor engine with mid turbione frame structure between turbine stages, establishing a nonlinear dynamic model of the dual-rotor system with mid turbine frame. Considering the nonlinear forces of the squeeze film damper and the rolling bearing, a test rig is established for experimental verification. The test results of the system vibration response show that the power turbine and the gas generator are coordinately connected with the mid turbine frame structure, and there is coupling between them, which makes the mid turbine frame structure dual rotor system produce coupling vibration.

Keywords: mid turbine frame; Coupling vibration; dual rotor ; basic vibration

1. Introduction

In recent years, mid turbine frame structures are widely used in advanced aero-engines. The mid turbine frame structure can effectively reduce the axial length of the engine, thereby reducing the overall weight of the engine and improving the efficiency of the engine [1]. The mid turbine frame structure carries the vibration between the gas generator rotors and power turbine simultaneously. The mid turbine frame structure carries the power turbine and the gas generator rotor with different speed at the same time, and the dynamic coupling phenomenon occurs between the rotors[2].The stiffness of the mid turbine frame structure will be weakened in the high temperature environment, which will adversely affect the rotor system and support structure of the engine. It is very important to study the dynamic coupling characteristics of the mid turbine frame structures.

Many researches have been made on the dynamic analysis of mid turbine frame structures. The vibration coupling problem of dual rotor system is usually the mutual excitation vibration coupling between rotor systems with mid turbine frame structure. The rotor has its own dynamic characteristics and produces complex vibration characteristics at the mid turbine frame. Tang Zhenhuan and Mi Dong took a type of dual rotor turboshaft engine as an example to establish a nonlinear dynamic model for a dual-rotor system. Through simulation and experimental verification, the results show that the angular stiffness of the mid turbine frame structure has a great influence on the coupled vibration, while the radial stiffness has almost no influence on the coupled vibration. By reducing the axial distance of the mounting side, the angular stiffness of the mid turbine frame structure can be increased, thereby weakening the coupled vibration of the system. [3]

Lei Binglong and Li Chao are aiming at the structural system of a high specific power turboshaft engine with mid turbine frame between turbine stages. They established the dynamic equation of the mid turbine frame rotor system to explore the system coupling vibration of the mid turbine frame structure between the power turbine rotor and the gas generator rotor. The calculation results show that the rotor fulcrum and the mid turbine frame structure are connected by force balance and displacement coordination, and there is stiffness coupling term. Chen Guo establishes a dual rotor – mid turbine farme dynamics model, the method of calculating the aviation engine vibration response, and the dynamic performance of the contra-rotating subsystem are analyzed, the results show that the aircraft engine right effectiveness of the whole machine vibration coupling dynamic modeling method.[5] Zhang Jian . proposed that the dynamic response of the rotor fulcrum has a certain influence on the dynamic stiffness characteristics of other rotors and the vibration response characteristics of the rotor, and the vibration influence of the dual rotors and the mid turbine frame structure should be considered in the calculation of the critical speed of the rotor.[6]

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2. Dynamic modeling

Simplified model based on turboshaft engine with mid turbine frame structure, the original engine is improved by using similarity design. Figure. 2 shows the test device of a dual rotor system with bearing mid turbine frame structure established by dynamic similarity design method. Figure. 3 shows the structure diagram of bearing mid turbine frame. The test rig rotor system throuth simulating the power turbine rotor, gas generator rotor and mid turbine frame. All fulcrums have rolling bearings and squirrel cage supports. There are SFDs on the 2nd, 3rd, 4th, and 6th fulcrums

Details of the mid turbine frame structure are shown in the enlarged section in Figure 3. Generally, turboshaft engine with a mid turbine frame will attempt to reduce the axial distance between the turbine casings and the mounting edge of the bearing web and the bearing drum and the mounting edge of the shared bearing seat. This structural form can greatly increase the angular stiffness at the mid turbine frame structure. In addition, since the load-bearing web is located behind the turbine of the gas generator, there is a large thermal deformation and thermal stress during operation, and the rigidity also changes to a certain extent. Therefore, the stiffness of the load-bearing drum is designed to be low, and the load-bearing drum is designed to have a lower stiffness. The deformation reduces the thermal stress of the load-bearing web, which also leads to a certain vibration coupling between the double rotors with the mid turbine frame structure. Vibration-coupled loads are generally transmitted by translational precession of the mid turbine frame transition. The difference between the dual-rotor system which have a mid turbine frame structure and the dual-rotor system have an intermediate bearing is that the vibration energy between the dual rotors with an intermediate bearing structure is transmitted through the intermediate bearing, and the transmission path is unique. The damping is only determined by the intermediate support structure; while the vibration energy between the two rotors with the mid turbine frame structure is transmitted through mid turbine frame structure, and can be transmitted through the two transmission paths of the bearing web of the mid turbine frame bearing seat and the gas generator. Multiple dampers and elastic supports can be arranged in the support bearing seat, and its stiffness and damping can be adjusted freely, which provides a large optimization design space for studying the vibration coupling of the system.

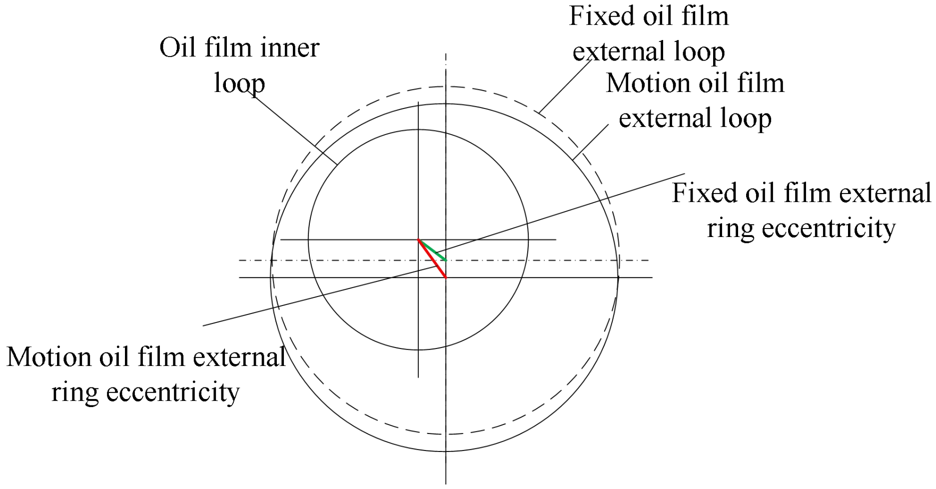

Figure1.SFD

The dual-rotor system with a mid turbine frame structure can transmit energy through the mid turbine frame structure, which can reduce the overall vibration of the system at a suitable rotational speed

The combination of SFD and rolling bearings in aero-engines and gas turbines can significantly reduce vibration. However, the vibration damping mechanism is quite complex and has not been fully elucidated yet. An important feature of the oil film force is nonlinearity, that is, the oil film force is a nonlinear force related to the eccentricity of the journal. The oil film pressure of the concentric SFD is always smaller than that of the non-concentric type, but its oil film force is different. The radial oil film molecules of concentric SFD is always larger than those of non-concentric SFD. The tangential oil film composition is the opposite. Therefore, the oil film

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stiffness of the concentric SFD is better, the damping capacity is worse than that of the non-concentric SFD, and the damping effect is worse.[7]

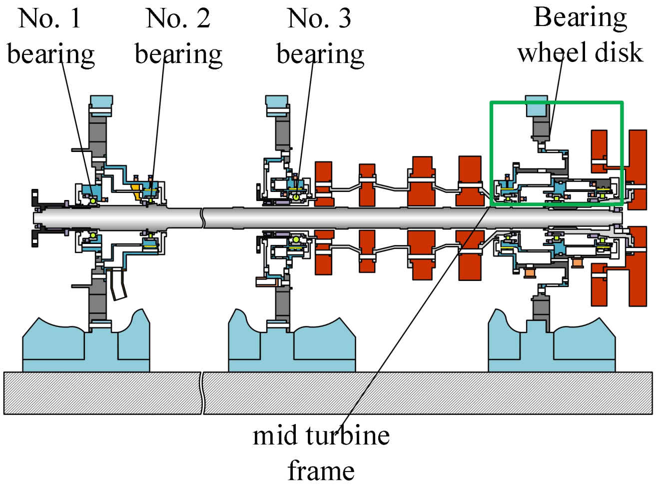

Figure2.Model of test rig

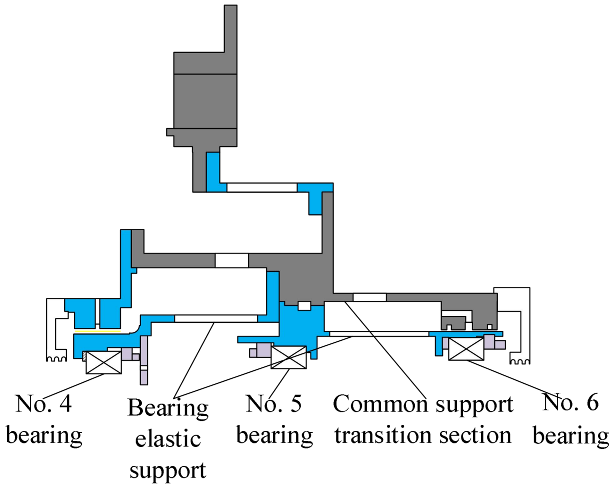

Figure3.Schematic diagram of bearing common cavity structure

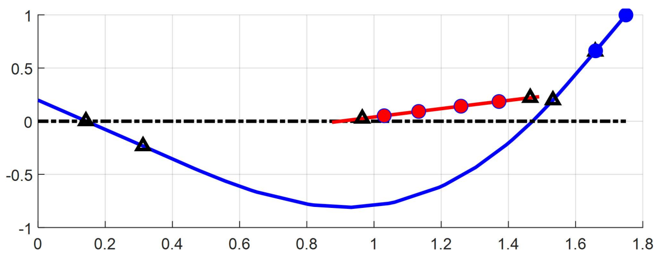

Figure4.vibration map

Through the simulation analysis, the mode shape diagram as shown in the above figure is obtained.The power turbine array is bending, and the gas generator is tilt-type.

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3. Parameters of test rig

3.1 SFD parameters

Table 1. SFD parameters

Position of SFD	No. 2 bearing	No. 3 bearing	No. 4 bearing	No. 6 bearing
Radius of the oil film (R/m)	65·10-6	67·10-6	69.5·10-6	66.8·10-6
The oil film length (L/mm)	23	28	19.2	28
The oil film length (C/m)	0.2·10-6	0.2·10-6	0.2·10-6	0.2·10-6

3.2 The bearing stiffness

Table 2. The bearing stiffness

Bearing number	The bearing stiffness(N/m)
K1	1·108
K2	3·106
K3	8.88·106
K4	8.9·106
K5	1·108
K6	3.5·106
KG	5.5·107

KG is the external supporting stiffness of the bearing common cavity

Figure5.Twin rotor test rig with mid turbine frame structure

The above photo is a dual rotor teste rig with a mid turbine frame. Rotated by two motors.

4. Experimental verification

Based on the rotor system model in figure. 2, a mid turbine frame and dual rotor system test rig is established, and the coupling vibration response test of the rotor is carried out with the test rig. Related Experimenter of the test rig are shown in the above table. and the figure.2 and figure.3.

4.1 Critical speed

Increase the speed of the gas generator rotor to 7000rpm. Keep the speed of the gas generator rotor constant, and gradually increase the power turbine rotor speed. The variation curves of amplitude and speed of power turbine rotor are shown in fig.5.

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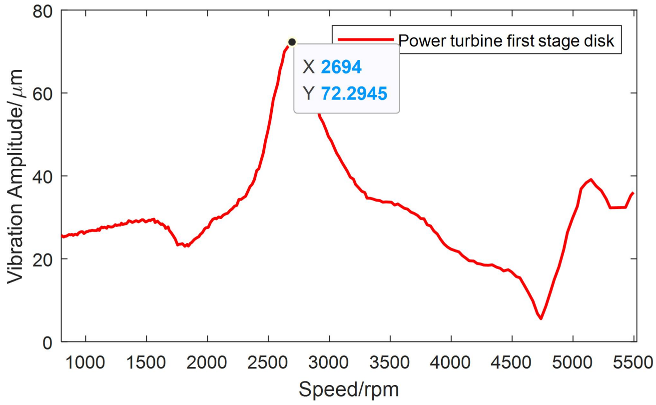

Figure6.Power turbine rotor amplitude curve

From figure 5 we can see that when the rotor speed of the gas generator is 7000rpm, the critical speed of the power turbine rotor is 2694rpm.

Change the speed of the gas generator and continue to measure the vibration response of the power turbine. Adjust the speed of the gas generator to 5000rpm and gradually increase the speed of the power turbine rotor to obtain the curve shown in figure. 6 below.

Change the gas generator speed, the power turbine rotor ‘s critical speed is 2494rpm. It can be seen that due to the change of the rotor speed of the gas generator, the critical speed of the power turbine rotor changes significantly, indicating that the rotor of the power turbine and the rotor of the gas generator generate vibration coupling by sharing the supporting structure.

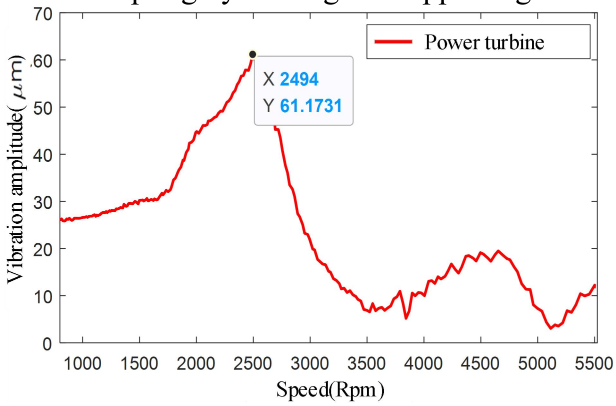

Figure7.Power turbine rotor amplitude curve

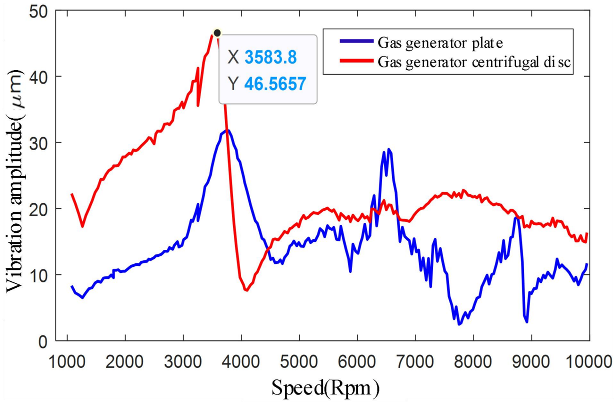

Figure8.Gas generator amplitude curve

The above figure shows the amplitude curve of the gas generator rotor. the power turbine rotor is at the critical speed. It can be seen from the figure that the critical speed of gas generator is 3584 rpm. The speed ratio about the gas generator to power turbine is 1.44. At this time, the system vibration is the largest, which is a dangerous speed.

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4.2 Frequency

Change the speed of power turbine rotor,.Keep the speed of gas generator constant at 5000rpm, measure the frequency octave data of gas generator rotor, and obtain the spectrum image as shown below .

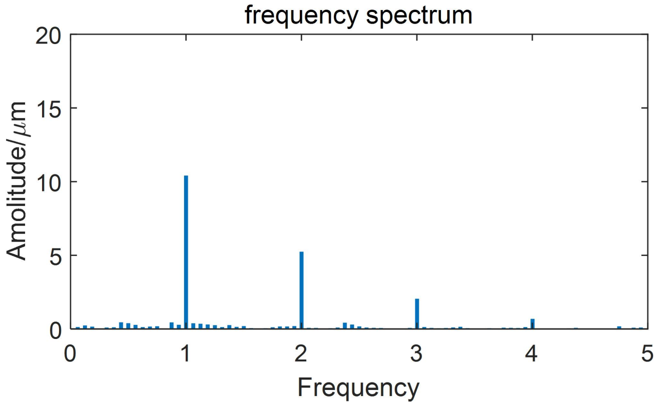

Figure9.The speed of power turbine is 0 rpm, and the speed of the gas generator is 5,000 rpm

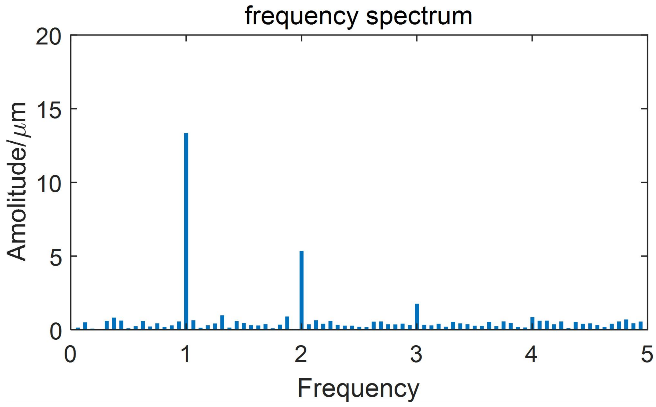

Figure10.The speed of power turbine is 1500 rpm, and speed of gas generator is 5,000 rpm

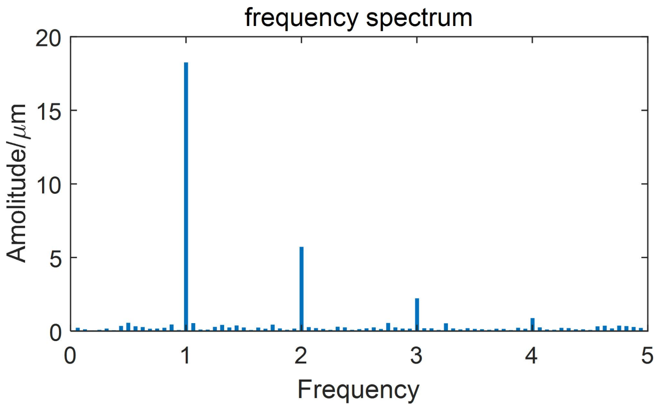

Figure11.The speed of power turbine is 2500 rpm, and the speed of gas generator is 5,000 rpm

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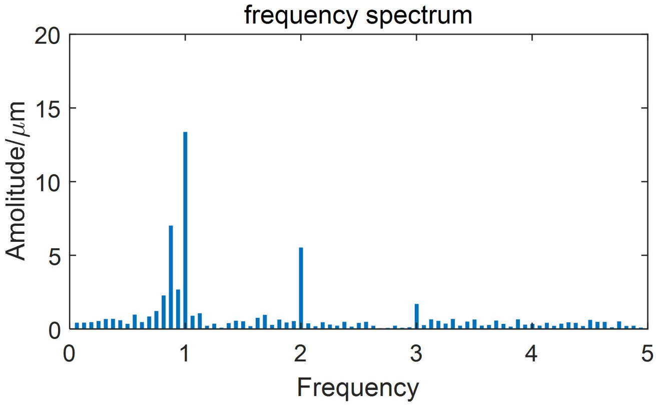

Figure12.The speed of power turbine is 4500 rpm, and the speed of gas generator is 5,000 rpm

When the rotor speed of gas generator is 5000rpm and the power turbine rotor is stationary, the one-frequency amplitude of the gas generator rotor is 11μm. Gradually increase the rotor speed of the power turbine to 1500rpm, at this time the upper frequency of the gas generator is 13μm. Continue to increase the rotor speed of the power turbine to 2500rpm. At this time, the rotor speed of the power turbine is close to the critical speed, and the vibration is relatively large. The first frequency on the gas generator rotor is 18μm. After the power turbine passes the critical speed, the doubling frequency on the gas generator decreases to 14μm at 4500rpm.

The above experimental data show that the speed of gas generator is 5000 rpm, and the amplitude frequency response of gas generator is detected by changing speed of power turbine. Through comparison, it can be seen that there is a relatively obvious cross excitation phenomenon on the rotor of the gas generator, and the unbalanced excitation of the power turbine can be clearly seen that in the amplitude frequency signal of the rotor of the gas generator. The unbalanced excitation of the power turbine can be clearly seen in the amplitude frequency signal of the gas generator rotor. And the greater the vibration of the power turbine rotor, the greater the amplitude of the first octave frequency on the gas generator.

4.3 Beat

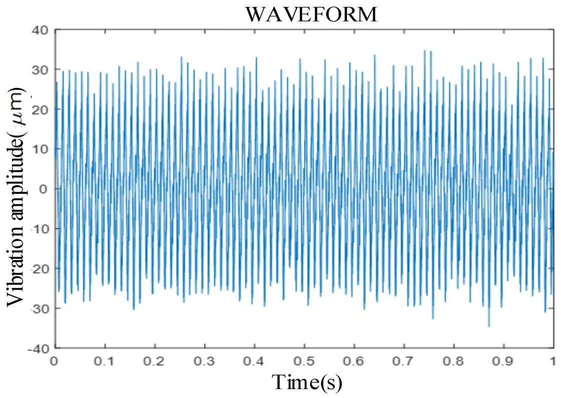

Figure13.Gas generator amplitude curve

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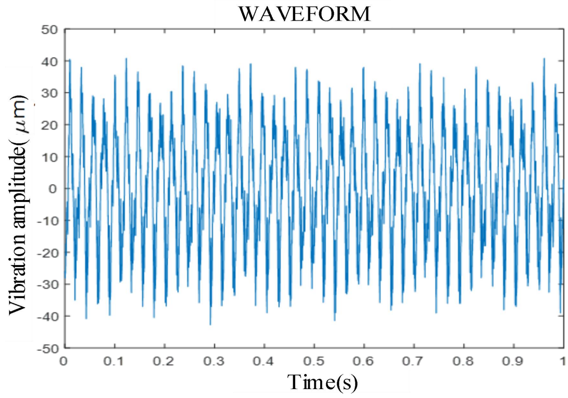

Figure14.Power turbine amplitude curve

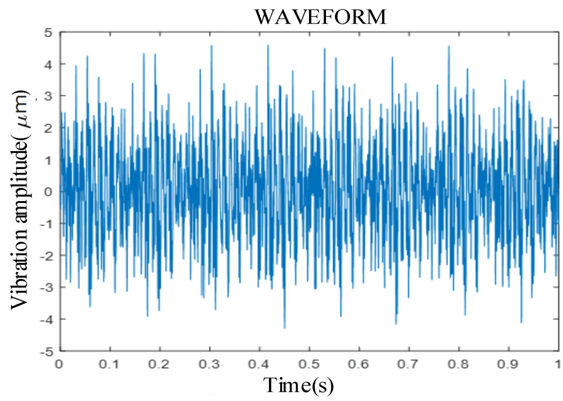

Figure15.Mid turbine frame amplitude curve

The rotor coupling vibration response test is conducted with this tester, and the test results are shown in the above figure:

The vibration phenomenon exists in both the power turbine and gas generator. And the vibration response of the mid turbine frame structure is the superposition of the rotor of power turbine and gas generator. Explain that power turbine and gas generator are coupled through the mid turbine frame structure.

5. conclusion

In this paper, the dual rotor system with mid turbine frame structure is taken as the research object, and the dynamic model is established through the rotor dynamics theory for simulation, and conduct model and whole machine testing. The main conclusions are as follows:

1. The vibration response of the dual rotor system with mid turbine frame structure consists of two parts, one is the vibration caused by the unbalanced excitation of the rotor itself, and the other is the vibration transmitted between the rotors through the mid turbine frame structure.
2. The experimental results show that there is vibration coupling in the min turbine frame dual-rotor system ,and the coupling characteristics are related to the rotor speed.
3. The experimental results show that the mid turbine frame structure of the bearing is reasonably designed and the vibration characteristics of the rotor are stable.
4. The speed of the gas generator and the speed of the power turbine should avoid the critical speed to avoid the harm of resonance to the system.

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