VIBRATION NUMERICAL ANALYSIS OF COUNTER-ROTATING TURBINE WITH WAKE-FLOW USING FLUID-STRUCTURE INTERACTION METHOD

Zhao Zhenhua[1], *Lü Wenliang*[2], *Chen Wei*[1], *Wu Tieying*[1]

(1. College of Energy and Power Engineering, NUAA, 29 Yudao Street, Nanjing, 210016, P. R. China;
2. The Third Academy, China Aerospace Science and Technology Corporation, Beijing, 100074, P. R. China)

Abstract: The design of counter-rotating turbine is one of new techniques to improve the thrust-weight ratio of jet propulsion engines. Numerical analysis of a low pressure (LP) counter-rotating turbine rotor blade is presented by using ANSYS/CFX software. Interaction of aerodynamics and solid mechanics coupling in the computation is applied. In some rating of turbine, stress distribution and vibration characteristics of low pressure turbine (LPT) blade are computed. The wake aerodynamic forces and LPT blade vibration are transformed in frequency domain using fast Fourier transform (FFT) method. The results show that under wake aerodynamic force excitation, the first order modal vibration is more easily aroused and the higher order response cannot be ignored. Moreover, with different temperature fields, the vibration responses of blade are also different.

Key words: counter-rotating turbine; fluid and structure coupling; vibration responses; numerical analysis

CLC number: V231.92; V232.4 **Document code**: A **Article ID**: 1005-1120(2011)01-0066-07

INTRODUCTION

The counter-rotating turbine is that, in a twin spool turbo-engine, two rotors rotate in the opposite directions, and there may be no nozzle guide vane (NGV) between the two rotors in order to save weight, to increase thrust-weight ratio and probably to improve engine performance. In recent years, great progresses have been made in technology of the counter-rotating turbine. Studies show that the use of counter-rotating turbine structure can reduce the number of components, lighten the structural quality, improve aerodynamic performance, reduce gyroscopic moment and so on. Counter-rotating turbines have great many advantages over conventional turbines[1-3].

However, upstream blades generate wake-flows that become periodical disturbance for downstream blades. When frequency matches natural frequency of blades, such pulses may excite vibration and causes high cycle fatigue[4] of the blades. In the other hand, low pressure (LP) blades of the turbine are relatively thin. At the high temperature, blade stiffness decreases and blade displacement may be significant enough to influence flow field. In this case, the research based on fluid and structure coupling is necessary. The analysis in this paper, based on ANSYS/CFX software, is in 3-D, unsteady and fluid-solid coupling numerical computation to analyze vibration response and stress field of low pressure turbine (LPT) blades as well as impact of temperature on the calculation results.

1 METHOD FOR FLUID-STRUCTURE COUPLING CACULATION

Fluid-solid coupling mechanics is a branch of the fluid and solid mechanics interaction, and many researchers have made great contributions in this area. A K Slone[5] solved the problem in the fluid-solid coupling of transient dynamics with

Received date: 2010-10-18; **revision received date**: 2011-01-06

E-mail: zhaozh27@yahoo.com.cn

the momentum conservation equation of the structure and the Euler equation of fluid mass, momentum and energy conservation. Zhang Qun and Toshiaki Hisada[6] proposed two kinds of methods for the fluid-solid coupling calculation. V Carstens[7] derived the control equations of gas-solid coupled system with the introduction of modal coordinates of the vector aeroelastic equations and 3-D N-S equations simultaneously.

There are generally two ways to solve the fluid-solid coupling problems. One is the alternate solution method, i. e., the weak-coupling method; another is the overall solution method, i. e., the strong-coupling method.

In this paper, the weak-coupling method is applied. The procedure can be described as follows: at moment t_i, firstly to solve the flow field, then to transfer the calculated pressure of the flow field into the solid model for the calculation of the solid domain to find displacement which modifies fluid grid and to re-compute flow field repeatedly using the new grid, and so on. Such a superposition, when the calculation reaches local convergence, the coupling calculation goes to the next step. Calculation process is shown in Fig. 1. After getting convergence at t_i, computation goes to next step for t_{i+1} until the final step.

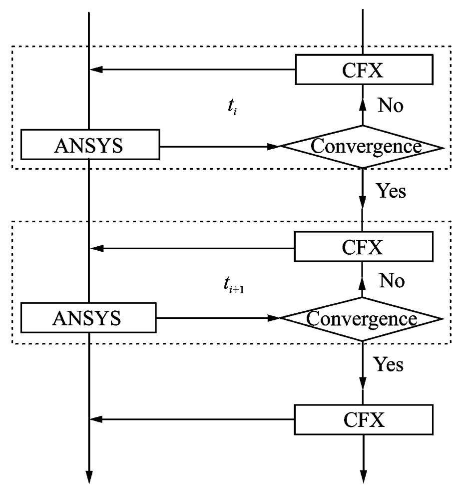

Fig. 1 Calculation process

2 NUMARICAL ANALYSIS

2.1 Physical model and boundary conditions

An finite element model (FEM) of a counter-rotating turbine is taken as an example. It has a high pressure (HP) NGV stator, a HP rotor and a LP rotor. There are 80 high pressure turbine (HPT) blades in HP rotor and 86 LPT blades in LP rotor (Fig. 2). Rotation speed is 14 978 r/min for HPT and 9 426 r/min for LPT. The study is focused on the LPT rotor blade, to analyze the structural strength and vibration properties in the design situation.

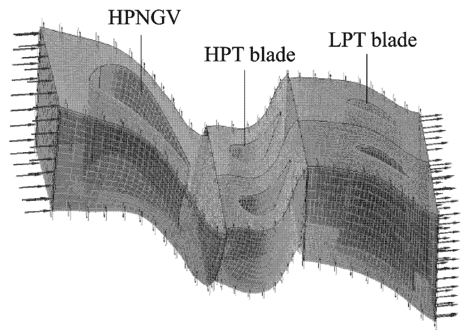

Fig. 2 Calculation model of CFX

At the inlet of HP NGV, total pressure is P_3^* and the distribution of total temperature vs blade height is shown in Fig. 3. The average outlet static pressure of LPT is P_4.

This unsteady coupling calculation takes ideal gas and unsteady Reynolds-averaged N-S equations as the control equations, using high-precision convection format. The second order backward Euler transient scheme is used in the transient format, and dynamic grid and k-ε model are used.

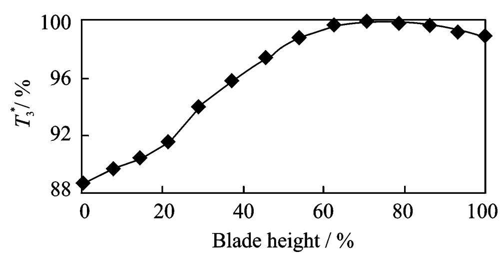

Fig. 3 Distribution of inlet temperature vs blade height

2.2 Calculation model

The CFX model of fluid calculation has 134 268 elements (Fig. 2). Transient coupling calculations and FEM of blade and bound form (the root of solid sticks) are shown in Fig. 4. It consists 3 648 elements and 18 984 nodes in total. The surfaces of LPT rotor blades are coupling surface, the coupling parameter is pneumatic pressure. The dynamic grid is applied in the region of the LPT rotor blades (Fig. 2).

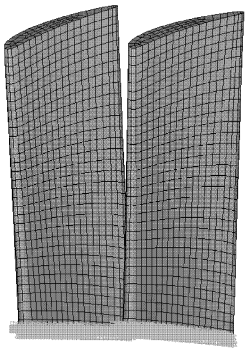

Fig. 4 FEM of LPT blade

2.3 Approach of temperature load

ANSYS/CFX does not provide function of coupling temperature between FEM and computational fluid dynamics (CFD) model. A program is developed to transfer the temperature from CFD model to FEM.

In one work state of the engine, the temperature in turbine is changing at the different moment since computation is unsteady. However, unsteady temperatures will not have a significant impact on turbine blades because of their thermal latency in the solid. In this case, temperatures in blades can be considered as steady. Considering the most dangerous situation, the temperature field at the moment of top temperature gradient will be exerted on the LPT rotor blade for analysis.

The approach steps are as follows:

(1) Using CFX to calculate unsteady temperature distributions of LPT rotor blade without deformation;

(2) Taking the one which has the greatest temperature gradient as example 1;

(3) Transforming this temperature field into the surface nodes of FEM by loading conversion developed by the authors;

(4) Computing the whole temperature distribution for all the nodes of FEM;

(5) Applying temperature results to FEM for further unsteady iteration analysis.

3-D linear interpolation method with the weighted average theory is applied to transform the temperature load. In Fig. 5, point P is a node of FEM and the grid is CFD model. Centered on P, a circle with r_0 is defined. Every node in CFD model in that circle is taken into account for weighted average for point P.

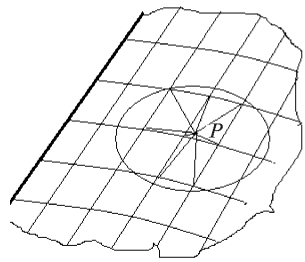

Fig. 5 Blade surface

The formula used in conversion program is

$$T_P = \begin{cases} T_i & i \text{ is } P \\ \dfrac{\sum \dfrac{T_i}{r_i}}{\sum \dfrac{1}{r_i}} & \text{Else} \end{cases}$$

where r_i is the distance between CFD node i and P, T_i the corresponding temperature in point i, and index i the CFD node number of points located in that circle. The program allows user to choose suitable r_0 to get more accurate interpolation results. In addition, this program can be used for other parameters such as pressure, and so on.

Basically, material properties vary with temperatures, like Young's module. These variations are considered in the computation.

2.4 Calculation result and analysis

Aerodynamic pressure, temperature and centrifugal load are acted on the turbine rotor blade. The LPT rotor blade is calculated used the fluid-solid coupling numerical calculation method under two kinds of blade temperatures (examples 1 and 2) in order to analyze forced vibrations of the LPT rotor blade under aerodynamic loads. The temperatures of the example 2 are globally proportionally lower than those of the example 1 considering the cooling effect.

Three points on pressure surface of the blade are selected for analysis, points A and B (Fig. 6) for stress response and points B and C for displacement response. Point A is near the hub of the blade and point C near the tip.

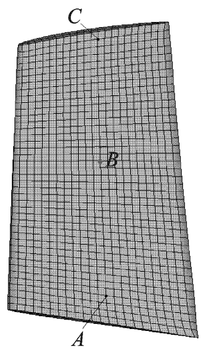

Fig. 6 Feature point position on LPT blade

Extracting the pressure fluctuation of the place approach point B (Fig. 7). In order to analyze the characteristics of wake-flow, the pressure curve is transformed in frequency domain using fast Fourier transform (FFT) method. Fluid excitation force shows broadband characteristics, and contains seven major frequency ranged from 6 006.5 Hz to 39 448.1 Hz. The frequency with maximum amplitude is 33 459.6 Hz, and frequencies 6 006.5, 21 500.7, and 27 489.2 Hz follow (Fig. 8).

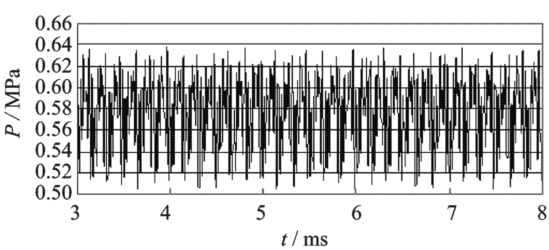

Fig. 7 Pressure curves of place approaching point B

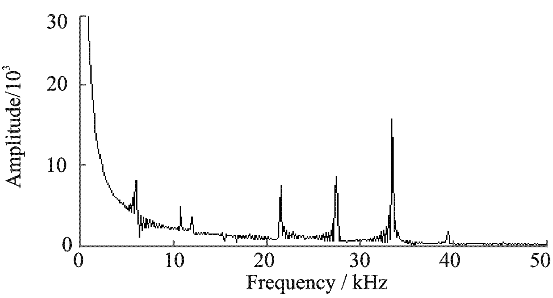

Fig. 8 Frequency spectrum of pressure at place approaching point B

Forced response of the LPT rotor blade is obtained with transient coupling method under different temperature conditions, the stress curves of points A and B are shown in Fig. 9 and the displacement curves of points B and C are shown in Fig. 10.

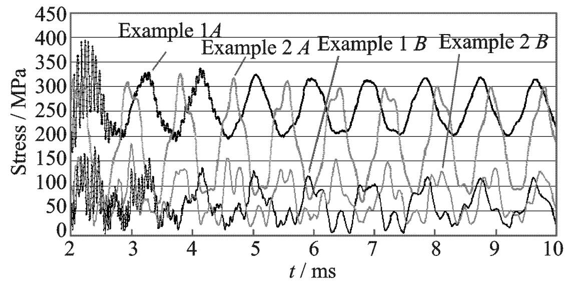

Fig. 9 Stress curves of points A and B under different temperature

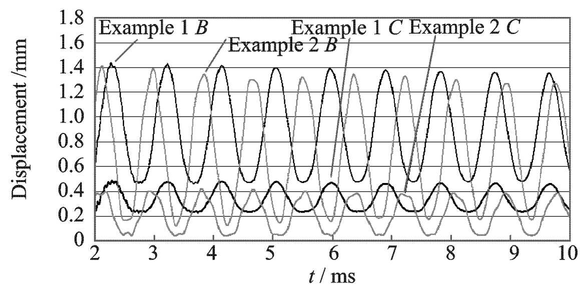

Fig. 10 Displacement curves of points B and C under different temperature

Form Fig. 9 and Fig. 10, we can find that: (1) The curves of the displacement are smoother than the curves of stress. Displacement and stress are not proportional. That means the vibration is not in its single natural modal. (2) Temperature has effect on the amplitude of stress response. The amplitude of stress response increases while the temperature decreases, but the mean stress increases with temperature. Here, the mean stress includes centrifugal effect and thermal stress because of temperature gradient in the blade. (3) The amplitude of displacement decreases when temperature increases, but the mean displacement increases.

3 SPECTRUM ANALYSIS OF BLADE FORCED RESPONSE

In order to analyze the forced response characteristics of LPT blade, the stress and displacement response signals obtained from the above computation of the selected points on the blade under different temperature field are transformed in frequency domain using FFT method. Spectrograms of the stress curves of points A, B and the displacement curves of points B, C of the examples 1 ,2 are shown in Figs. 11-14. The frequency statistics of the selected points is listed in Table 1.

Table 1 Frequency statistics under different temperature fields Hz

Parameter	Example 1				Example 2			
	a	b	c	d	a	b	c	d
Stress of point A	1 098	2 196	3 418	16 855	1 222	2 321	3 760	18 531
Stress of point B	1 099	3 420	16 432	—	1 221	4 511	5 610	—
Displacement of point B	1 097	2 201	3 470	16 381	1 220	3 789	4 516	—
Displacement of point C	1 100	2 201	3 196	16 761	1 222	4 516	—	—

The frequencies of point a in the examples 1, 2 are the first natural frequency of the blade under the respectively state (Fig. 11). The frequencies of point d in Fig. 11(a), point c in Fig. 12 (a), point d in Fig. 13(a) and point d in Fig. 14 (a) are the seventh natural frequency of blade. The frequencies of point b in Fig. 12(b), point c in Fig. 13(b), and point b in Fig. 14(b) are the second natural frequency of the blade. 2 196, 2 201 and 3 418, 3 420, 3 196 Hz in example 1 are frequency-doubled and tripled harmonic. 2 321 and 3 760, 3 789 Hz in example 2 are also frequency-doubled and tripled harmonic.

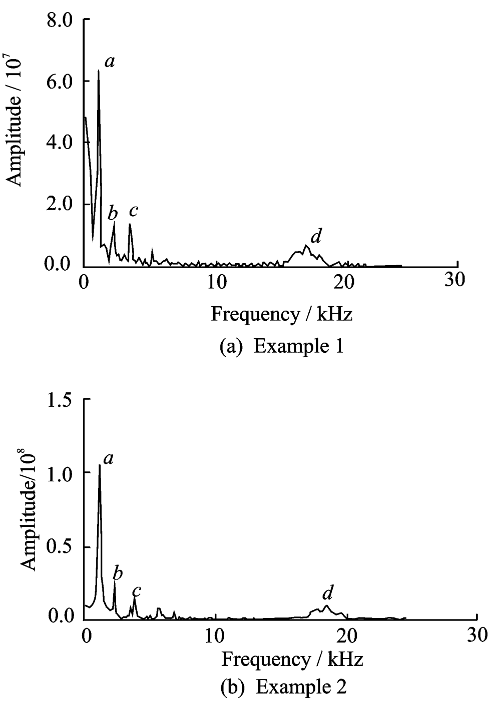

Fig. 11 Frequency spectrum of stress at point A under different temperature

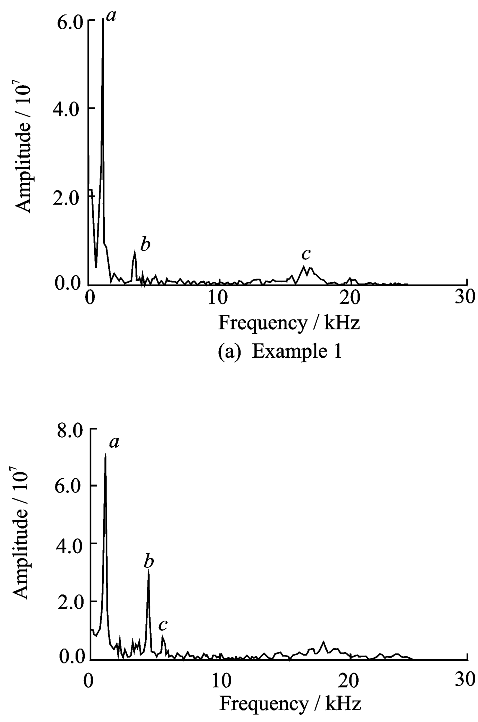

Fig. 12 Frequency spectrum of stress at point B under different temperature

In summary, the forced response of the LPT rotor blade in example 1 is mainly composed by the first and the seventh natural modes. It can be seen through the frequency spectrum that the first natural mode dominates the response. The forced response in example 2 is composed by the first and the second natural modes, and the first mode dominates the response too.

Comparison of the frequency characteristics of wake aerodynamic force and blade vibration shows that: (1) Due to the wide frequency band

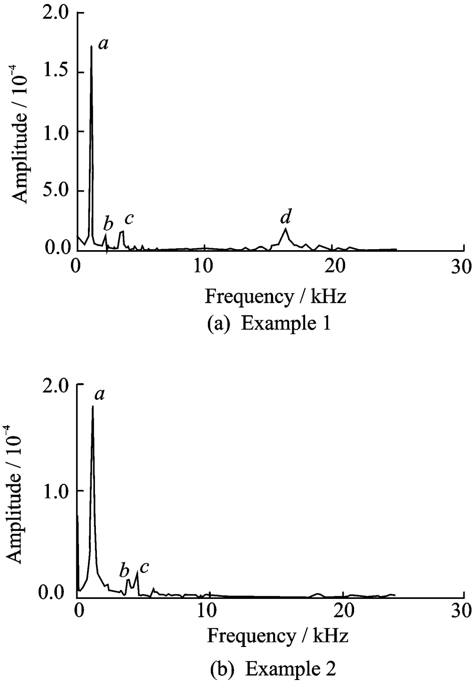

Fig. 13 Frequency spectrum of displacement at point B under different temperature

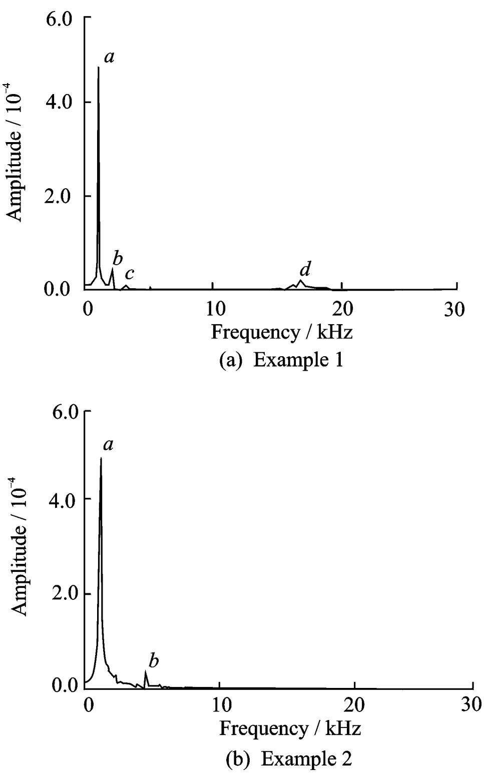

Fig. 14 Frequency spectrum of displacement at point C under different temperature

of exciting force with high energy, the first order vibration modal is easily aroused. (2) The higher order modals may also appear and they are not ignored when the amplitude is big enough.

4 CONCLUSIONS

By using ANSYS/CFX software, the process of computation of fluid-structure coupling is demonstrated successfully. Through this process the counter-rotating turbine is analyzed aerodynamically and structurally. After temperature conversion, vibration is analyzed in the LPT blade under two different temperature distributions. Based on above results, the main conclusions are:

(1) The curves of the displacement are smoother than that of stress. They are not proportional. That means the vibration is not in single natural mode in this operation rating of the turbine.

(2) Under wake aerodynamic force excitation, the first order modal vibration is more easily aroused and high order response cannot be ignored.

(3) With different temperature fields, natural modals of the blade are different and vibration responses are also different.

(4) The blade vibration response under wake-flow excitation conditions is complex and needs further studies.

References:

[1] Louis J F. Axial flow contra-rotating turbines[R]. ASME 85-GB-218, 1985.

[2] Liu Siyong, Wang Ping, Fang Xiangjun, et al. Research on aerodynamics design of counter-rotating turbine without intermediate vanes[J]. Gas Turbine Exprement and Research, 2002, 15(1): 20-27. (in Chinese)

[3] Ji Lucheng. Review and prospect on research of counter-rotating turbine[J]. Aeroengine, 2006, 32 (4): 49-53. (in Chinese)

[4] Ji Lucheng. Analysis for technique challenges on counter-rotating turbomachinery [J]. Journal of Propulsion Technology, 2007, 28 (1): 40-44. (in Chinese)

[5] Slone A K. Dynamic fluid-structure interaction using finite volume unstructured mesh procedures [J]. Computer and Structures, 2002, 80: 371-390.

[6] Zhang Qun, Hisada Toshiaki. Studies of the strong

coupling and weak coupling methods in FSI analysis [J]. Int J Numer Meth Enging, 2004(60): 2013-2029.

[7] Carstens V, Kemme R, Schmitts S. Coupled simulation of fluid-structure interaction in turbo machinery [J]. Aerospace and Technology, 2003, 7: 298-306.

尾流作用下对转涡轮叶片振动的流固耦合数值分析

赵振华[1] 吕文亮[2] 陈 伟[1] 吴铁鹰[1]

(1.南京航空航天大学能源与动力学院,南京,210016,中国;2.中国航天科工集团公司三院,北京,100074,中国)

摘要:采用对转涡轮设计可以提高喷气发动机的推重比。本文应用ANSYS/CFX软件,采用流固耦合数值分析方法对低压涡轮转子叶片进行了分析。对一定的涡轮工况,得到了高压涡轮转子叶片尾流作用下的低压涡轮转子叶片振动的应力和变形变化规律。采用傅里叶变换,对流体激励力、叶片应力及变形相应进行频谱分析,结果表明:在复杂来流激励下,叶片的第一阶振动较易被激起;但高阶振动响应不可忽略;而且叶身温度场不同,被激起的振动阶次也不同。

关键词:对转涡轮;流固耦合;振动响应;数值分析

中图分类号:V231.92;V232.4

(Executive editor: Sun Jing)