INVESTIGATION ON THE UNSTEADY PRESSURE FLUCTUATION CHARACTERISTIC IN THE BLADE TIP SEAL OF STEAM TURBINE BASED ON SPECTRUM

by

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The paper presents the unsteady numerical simulation results of tip leakage flow in high pressure steam turbines, and also presents the influence analysis of leakage vortexes on pressure fluctuation characteristics of rotor cascade under different blade tip seal clearances. The numerical method for calculating is based on the large eddy simulation turbulence model and the pressure fluctuation of rotor cascade which induced by the unstable leakage vortexes is obtained by frequency spectrum analysis. The results show that the vortex frequencies in tip seal cavity contain both the wheel rotating frequency and the high frequency caused by the tip leakage flow breaking into small scale vortexes. The unsteady characteristics of tip leakage flow also induce steam exciting force which changes with the time.

Key words: *steam turbine, tip leakage vortex, pressure fluctuation, steam exciting force, large eddy simulation*

Introduction

There is an inevitable tip clearance at the blade tip in axial steam turbine. Part of steam cross over the blade tip from the pressure side to the suction side and form the leakage losses. The leakage losses, which could contribute to roughly as much as 30% of the total losses in a stage, are the result of leakage vortex motion. The leakage vortex motion is affected by rotation speed of rotor, rotor-stator interaction, stator wake effects and their interactions, so the leakage flow in the blade tip clearance is complicated 3-D unsteady flow, which induces unsteady pressure pulsation at the blade tip. The steam exciting force in the blade tip clearance is fluctuated with the pulsation of pressure. Therefore, the flow characteristics of tip leakage flow and spatial-temporal evolution of leakage vortex should be analyzed in order to obtain the law and the circumferential propagation characteristics of pressure pulsation at the blade tip under different clearances. This study plays an important role to avoiding the rotors instability and improving the operation stability of steam turbine.

In recent years, detailed reviews of the complicated flow fields in blade tip clearances have been provided by a number of authors [1-7], shown that there were several vortex structures in the blade tip seal, including the vortex in the inlet cavity and outlet cavity of integral tip shroud, the vortex in the seal cavity as well as the mixing vortexes of the leakage flow with the main flow. Wei *et al.* [8] researched the influence of blade tip seal structure on the status of leakage vortex motion in the inlet cavity and outlet cavity of seal by the perspective way.

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Pfau *et al.* [9] carried out the experimental analysis on the interaction between the leakage flow and main flow in the passage of two-stage turbine, and attempted to look for the approaches reducing leakage losses and mixing losses. Gao *et al.* [10] studied the influence of tip leakage flow on main flow, and suggested that the ring vortex in the honeycomb seal can significantly reduce the circumferential speed of leakage flow in the seal cavity, so the mixing losses is reduced. Sun *et al.* [11] researched the leakage flow at different tip clearances by unsteady numerical simulation method, and found that the unsteady fluctuation of leakage flow on the rotor tip would be more intense under large clearance.

The aforementioned numerical simulations for turbulence in the blade tip clearance were mainly carried out by solving Reynolds-averaged N-S equations (RANS). Therefore, there are some limitations for solving unsteady strongly rotational turbulence in the steam turbine stage. Compared with RANS, more wealth information about unsteady flow field can be obtained by large eddy simulation (LES) that looks at the unsteady turbulence problem from the perspective of vortex system and takes sufficient account into the non-linear influence of micro-vortex on large vortex. You *et al.* [12] pointed out that there was a strong circumferential velocity component and leakage vortex separation in the flow field of blade tip clearance in axial steam turbine through LES, and the leakage vortex became larger with the leakage flow developing downstream and form intense turbulent fluctuation at the trailing edge of rotor blade. Cao *et al.* [13] obtained more detailed temporal and spatial change features and vorticity distribution of leakage flow in blade tip clearance of steam turbine. However, none of these researches further explained unsteady pressure fluctuation at blade tip.

Test measurement is the direct way for acquiring information about pressure fluctuation. The experimental study was carried out usually in the stage environment of staggered blade cascade [14]. Liu *et al.* [15] measured the pressure fluctuation on the rotor surface at blade tip clearance by the dynamic pressure sensor on the end wall of single row rotors, and obtained the relation between the dominant frequency of pressure fluctuation and the rotation speed. Ling *et al.* [16] measured the unsteady features of vortex in the separated flow on the surface of blades at plane cascade tester, and experimental studied on the influence of external drive on the performance of blade cascades. However, as restrained by environmental and experimental conditions, it is technically difficult to measure the pressure fluctuation in the seal clearance at blade tip of steam turbine. With the development of CFD technology, numerical simulation has become an important means for researching the features of pressure fluctuation of unsteady flow field in seal clearance at the blade tip.

The paper conducted unsteady computation on the blade tip leakage flow under different seal clearances by means of LES, and analyzed the characteristics of pressure pulsation on the surface of rotor cascade shroud, so as to obtain the change of pressure pulsation on the cascade surface under different seal clearances. Finally, this paper discussed the distribution law of steam exciting force at the blade tip seal clearance induced by leakage vortexes.

The CFD Modeling and simulation

Subgrid-scale stress model

In LES, the instant continuity equation and the momentum equation can be obtained by the spatial filtering of control equations and be used for solving large scale vortex flow field. After filtering, momentum transport between small scale vortex and large-scale vortex can be reflected in subgrid-scale stress, its mathematical model can be indicated as the equation:

$$\frac{\partial}{\partial t}\left(\rho \bar{u}_i\right)+\frac{\partial}{\partial x_j}\left(\rho \bar{u}_i \bar{u}_j\right)=-\frac{\partial \bar{p}}{\partial x_i}+\frac{\partial}{\partial x_j}\left(\mu \frac{\partial \bar{u}_i}{\partial x_j}\right)-\frac{\partial \tau_{ij}}{\partial x_j} \tag{1}$$

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x_i}\left(\rho \bar{u}_i\right)=0 \tag{2}$$

where $\bar{u}$ is the average speed of fluid, ρ – the density of fluid, $\bar{p}$ – the time average pressure, μ – the dynamic viscosity, i and j are the tensor indicators, i, j = 1, 2, 3, and the τ_{ij} is in the following forms:

$$\tau_{ij}=\rho \overline{u_i u_j}-\rho \overline{u_i}\,\overline{u_j} \tag{3}$$

Smagorinsky-Lily subgrid-scale stress model can correctly forecast the generation and dissipation of small-scale vortex, which is verified in the flow model of turbomachinery [17]. Therefore, this paper selects Smagorinsky-Lily subgrid-scale stress model as the SGS model for LES of the turbulent flow in the tip seal clearances of steam turbine. Assume that the subgrid-scale stress is in the following forms:

$$\tau_{ij}-\frac{1}{3}\tau_{kk}\delta_{ij}=-2\nu_t \bar{S}_{ij} \tag{4}$$

$$\nu_t=\left(C_s \Delta\right)^2\left|\bar{S}\right| \tag{5}$$

$$\left|\bar{S}\right|=\sqrt{2\bar{S}_{ij}\bar{S}_{ij}} \tag{6}$$

$$\bar{S}_{ij}=\frac{1}{2}\left(\frac{\partial \bar{u}_i}{\partial x_j}+\frac{\partial \bar{u}_j}{\partial x_i}\right) \tag{7}$$

where ν_t is the turbulent viscosity of subgrid-scale stress model, τ_{kk} refers to one of the subgrid-scale isotropic, $\bar{S}_{ij}$ – the rate of stress tensor, Δ – the filtering scales, C_s – the Smagorinsky constant, in order to make the balance of subgrid-scale of turbulent kinetic energy dissipation and large scale turbulent kinetic energy, this paper selects C_s = 0.1.

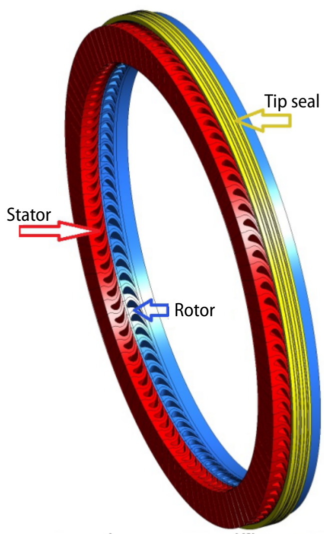

Figure 1. Shrouded turbine geometry

Calculation model and boundary conditions

The flow passage of second high pressure stage of steam turbine was performed as a computational fluid domain in the 3-D computational investigation. The extension of stator inlet, the extension of rotor outlet, the stator passage, the rotor passage and the blade tip seal section which was included in the investigation were given in fig. 1. In order to analysis the characteristics of pressure pulsation in blade tip seal cavities under different clearances, three different models of blade tip clearances were constructed with the clearance 0.5 mm, 0.75 mm, and 1 mm, respectively.

As shown in fig. 2, in view of the large calculation model, seriously distorted stator and rotor flow passages, complicated structure in tip seal clearances, and in order to facilitate calculation, multi-block structured hexahedral mesh was used for dividing the calculation region into grids, and dividing fine grids at the tip seal clearances which were the emphasis of analysis. The independence of the grids was verified by using two important parameters, *i. e.*, isentropic efficiency and average pressure on shroud surface. As shown in fig. 3, when the number of the grids reached 12 million, the parameters monitored were no longer sensitive to calculation if the number of grids were increased. Therefore, the total numbers of grids were controlled within 12.5 to 14 million.

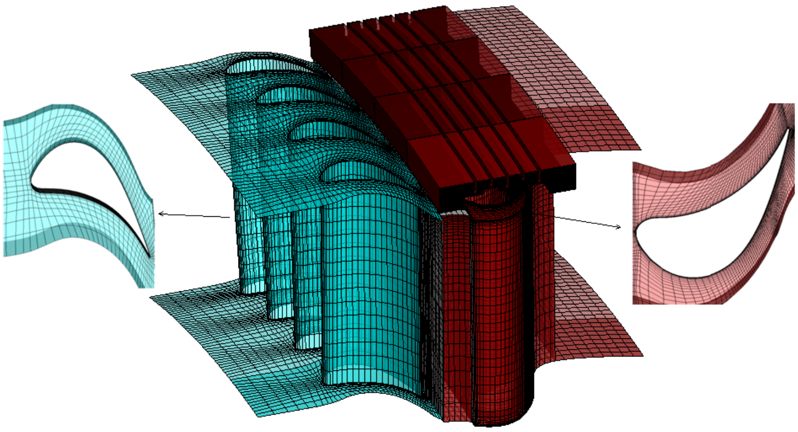

Figure 2. Computational grids

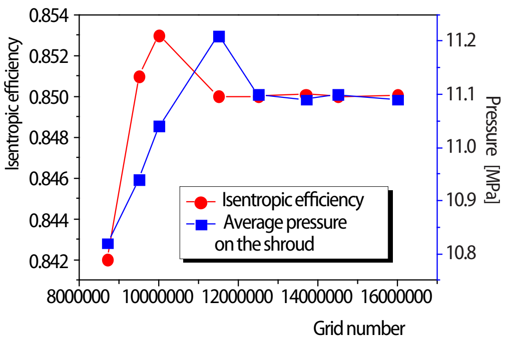

Figure 3. Dependence verification of grid number

Unsteady flow field was solved by using LES, and second-order upwind scheme was selected for discretization. The working medium was superheated steam. The inlet and outlet of the model was assumed as the pressure-inlet and the pressure-outlet. Setting of boundary parameters: total pressure at the inlet of 11.77 MPa, temperature of 756 K, static pressure at the outlet of 10.9 MPa, temperature of 734 K. The interface between the flow fields of rotor and stator as well as between rotor and seal were disposed by using sliding mesh. The convergence results from steady calculation were used as the initial value for calculation of large eddy simulation to carry out iteration, which could obtain steadier condition of convergence from the simulation calculation.

Simulation Results and Analysis

Distribution of circumferential pressure in blade tip seal

Figure 4 shows the relative locations between rotor and stator at several moments in a rotation period. The symbol of $0/4T$, $1/4T$, $2/4T$, $3/4T$ and $4/4T$ in the figure represent the several moments in a period, respectively. It is regulated that the time when the rotor blade turn around a passage of stator cascade is a complete period T, because the time when rotor make a full rotation is 0.02 seconds and there are 100 passages in the stator cascade, so a period T is about 0.0002 seconds. According to fig. 4, at t = 0/4T moment, the leading edge of rotor blade is located below the trailing edge of stator blade, at t = 4/4T moment, the leading edge of rotor blade is just moving below the trailing edge of adjacent stator, and at t = 1/4T, t = 2/4T, and t = 3/4T moments, the leading edge of rotor blade is just located in the 25%, 50%, 75% of stator pitch, respectively.

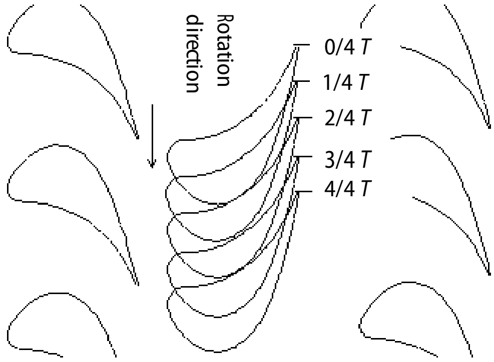

Figure 4. Relative position of rotor and stator

Through the simulation calculation, it is found that the variation of circumferential pressure in the blade tip seal is essentially the same under three different tip clearances. Taking the calculation with 0.5 mm high clearance as a research object, fig. 5 shows the distribution of circumferential pressure in the inlet, the outlet, the tooth tip and the cavity of labyrinth seal. It can be seen that the variation trend of circumferential pressure in the inlet of labyrinth seal is obvious different from the other three positions with the time changing. The distribution

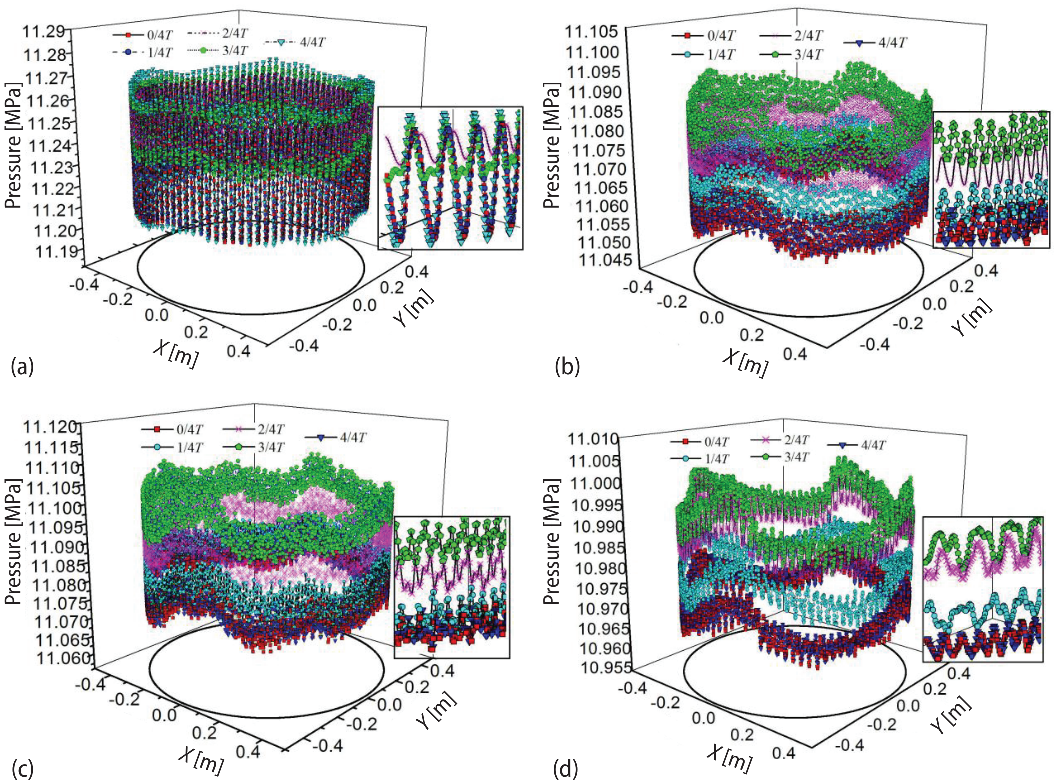

Figure 5. Distribution of circumferential pressure in tip seal clearance; (a) in the inlet of labyrinth seal, (b) at the tip of labyrinth seal tooth, (c) in the cavity of labyrinth seal, and (d) in the outlet of labyrinth seal

of circumferential pressure in the inlet of labyrinth seal presents strong and weak alternately at the $0/4T$ and $1/4T$ moments. The low pressure area is due to the spread of stator wake to the inlet of labyrinth seal. So, the number of stator wake is corresponding to the number of pressure change. The strength of circumferential pressure which changes from weak to strong and then from strong to weak is weakened at the $2/4T$ and $3/4T$ moments. The pressure of low pressure area which induced by the stator wake is obvious enhanced. This is because that the movement of the high-speed rotating cascade changes the pressure field between stator and rotor. The steam occur an extremely short retention when the leading edge of rotor blade passes by, reducing the speed and increasing the pressure of steam, so there is a higher pressure at the leading edge of rotor blade. In contrast, when the rotor passage passes by, the steam move into the rotor passage quickly, the pressure is reduced. The leading edge of the rotor just passes by the stator wake area between the $2/4T$ and $3/4T$ moments. Since there is a higher pressure at the leading edge of rotor blade, the drop rate of pressure in low pressure area which induced by the stator wake is the minimum at the two moments. When the leading edge of rotor blade leaved from the low pressure area, at the $4/4T$ time moment, the original state of low pressure is restored, so the strength of circumferential pressure distribution which presents strong and weak alternately is increased rapidly. It can be seen from the distribution of circumferential pressure that the stator wake spread and the rotor-stator interaction are the main cause of the uneven circumferential pressure in the inlet of labyrinth seal.

Figures 5(b) and 5(c) show the distribution of circumferential pressure at the tip of labyrinth seal tooth and in the cavity of labyrinth seal, respectively. We can see that the circumferential pressure presents no longer strong and weak alternately, but presents a finely zonal distribution. This is because there are spiral circumferential vortex movements in the cavity of seal. Those vortices impact the seal teeth and end wall, form many small-scale vortexes, and cause the circumferential pressure extremely complex in the labyrinth seal. But the largest pressure differential of circumferential pressure distribution is 0.0075 MPa at any moment, which is greatly reduced comparing with the inlet of labyrinth seal. This shows that the stator wake will be weakened gradually when the leakage flow is in spread of blade tip seal clearance. In addition, due to the spread of pressure field between stator and rotor to the downstream of the tip seal, the circumferential pressure fluctuates at different moment of a period in the tooth tip and the cavity of labyrinth seal. The average pressure is the highest at $3/4T$ moments, and is the minimum at $0/4T$ and $4/4T$ moments in the diagram. Thus, it can be seen that the complicated vortex structure of the tip seal leakage flow and the spread of alternately changing pressure field between stator and rotor in the tip seal clearance are the main cause of the uneven circumferential pressure in the tip seal cavity.

Figure 5(d) shows the circumferential pressure distribution in the outlet of labyrinth seal. Without considering the interaction between rotor and next stator cascade, the circumferential pressure returns to the state of inlet, namely the shroud should also bear the strength changes of circumferential pressure with the same frequency of stator wake. But the strength of circumferential pressure distribution which changes from weak to strong and then from strong to weak is weakened, basically the same as in the tooth tip and the cavity of labyrinth seal. The reason is that the leakage vortex in seal cavity influences on the circumferential pressure in the spreading process.

Pressure fluctuations of pressure monitoring points in tip seal clearance

In order to analyze the development of the unsteady pressure pulsation of the leakage vortex in blade tip seal clearance over time, pressure monitoring points were uniform arranged in the same circumferential position of blade tip seal clearance along the axial flow to gather the unsteady signal of pressure. The pressure pulsation coefficient, C_p, is used to show the pressure fluctuation signal at different locations of the blade tip seal:

$$C_p = \frac{p - \overline{p}}{0.5\rho U_{tip}^2} \tag{8}$$

where p is the instantaneous pressure, $\overline{p}$ – the average time pressure in calculation, ρ – the steam density in inlet of rotor passage, and U_{tip} – the circumferential speed of rotor tip.

Figure 6 shows the distribution features of the coefficients at each position of the blade tip under different clearances at different moments. The horizontal ordinate indicates that the sampling time is the seven-rotating period of blades. It shows that the pressure waveform of the inlet, fig. 6(a), and outlet, fig. 6(d), of the seal clearance are similar to irregular sine curve, which indicates that the pressure fluctuation of the inlet cavity and outlet cavity of the seal present a periodicity which is basically synchronous with the rotation of blades, *i. e.*, the rotational motion of the blades causes the pressure pulsation at the two eddies. In addition, compared with the pressure fluctuation in the outlet of labyrinth seal, it does not present complete periodicity in the inlet of labyrinth seal. This indicates that the vortex motion in the inlet

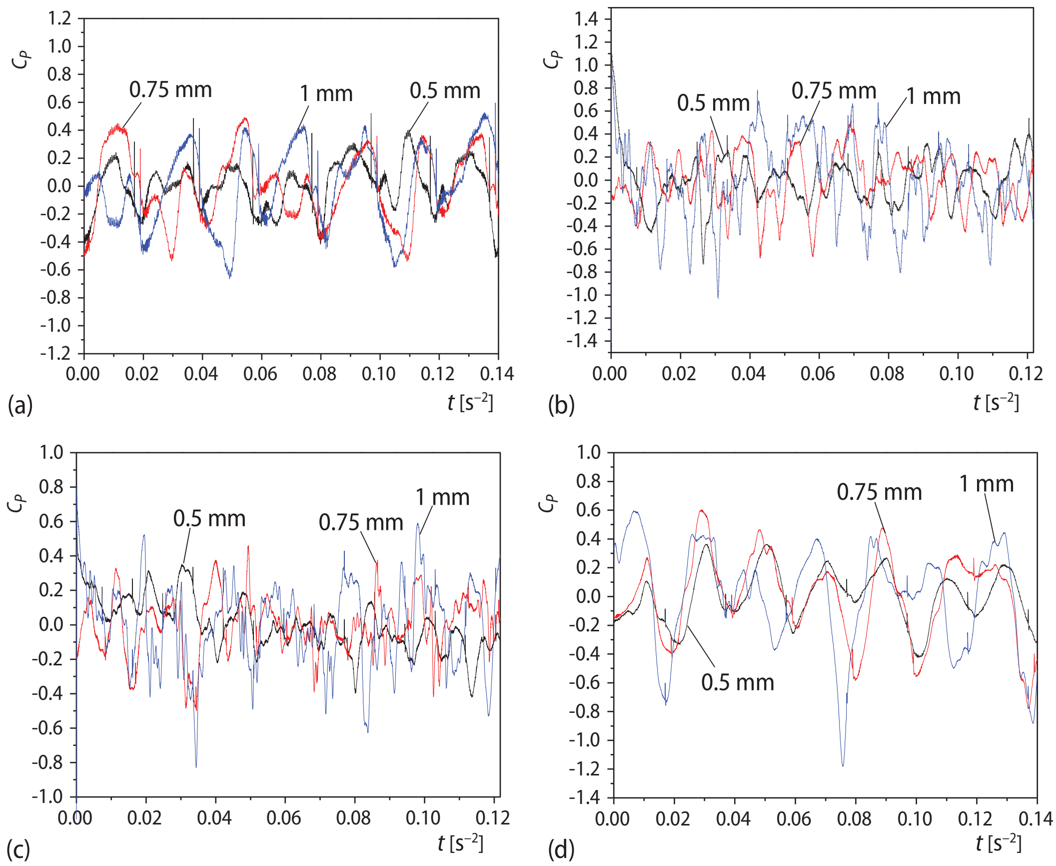

Figure 6. Pressure pulsation of monitor point in tip seal clearance; (a) in the inlet of labyrinth seal, (b) at the tip of labyrinth seal tooth, (c) in the cavity of labyrinth seal, and (d) in the outlet of labyrinth seal

of labyrinth seal has some particularity due to the influence of prewhirl effect and wake vortex spread of the stator. With the increase of blade tip clearance, the amplitude of pressure fluctuation increases gradually. This indicates that when the blade tip clearance increases, the energy will increase at the tip leakage flow, which enhances the change of leakage vortex.

According to figs. 6(b) and 6(c), under LES, complicated fluctuation exists in the tooth tip and the cavity of labyrinth seal. The periodical law is not obvious. This indicates that the rotational motion of blade cascades is not the only factor that affects external pressure pulsation at the external zones at these two positions. The tip leakage vortex is decomposed and collapsed in the clearance and cavity, which has an obvious influence on the pressure pulsation at these two places. In addition, with the increase of blade tip clearance, the pressure fluctuation in the clearance and cavity also increase, which indicates that the stability of the tip leakage vortex decreases with the clearance increasing.

Taking the monitoring results as the base parameters for the fast Fourier transform analysis, fig. 7 shows the frequency spectrum analysis of pressure at the monitoring points.

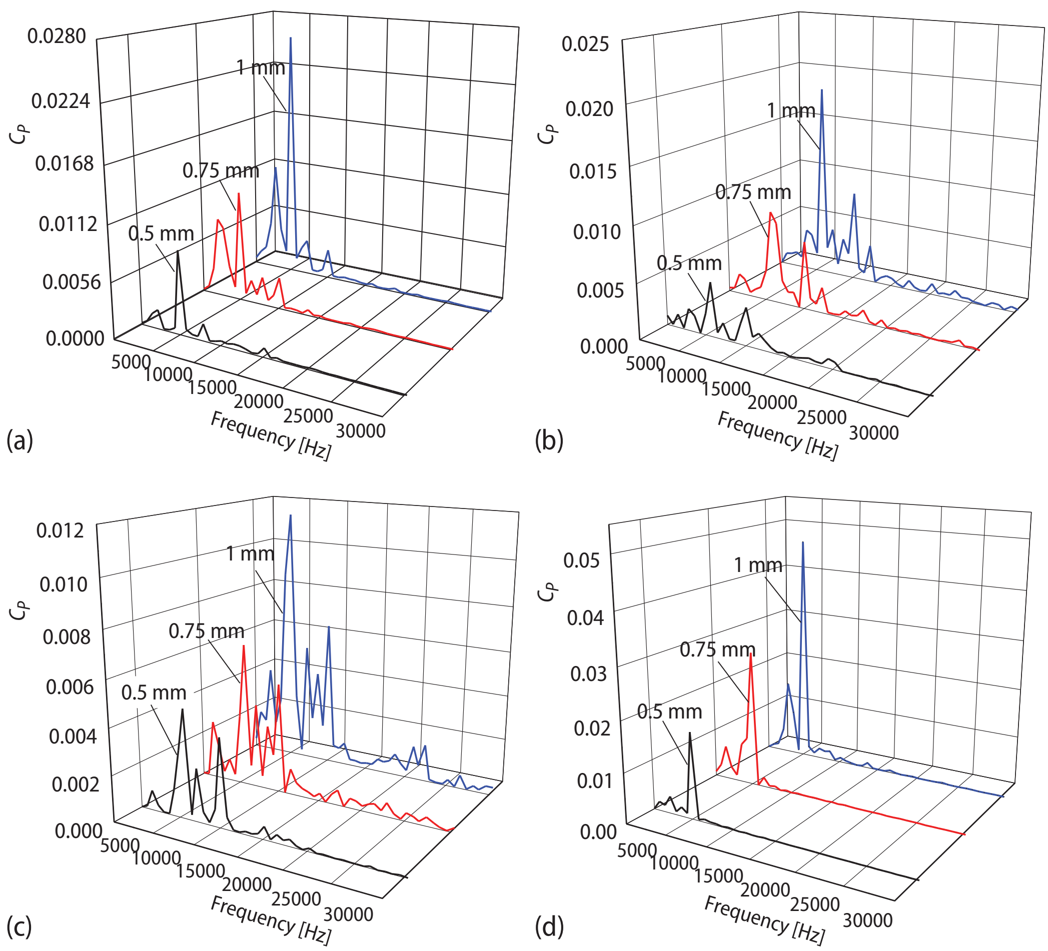

Figure 7. Frequency analysis of pressure in tip seal clearance; (a) in the inlet of labyrinth seal, (b) at the tip of labyrinth seal tooth, (c) in the cavity of labyrinth seal, and (d) in the outlet of labyrinth seal

According to the figure, in this case, the frequencies of the monitoring points have the same fundamental frequency marked 5000 Hz. And the frequency of the blade rotation calculated by the eq. (9) is also 5000 Hz:

$$f = \frac{Zn}{60} \tag{9}$$

where n is the rotation speed, $n = 3000$, and Z is the stator blade number, $Z = 100$.

This shows that the pressure in tip seal cavity present periodic fluctuations in general. With the increase of blade tip clearance, the amplitude of frequency increases. This phenomenon is corresponding to the pressure fluctuation of fig. 6. The frequency components of pressure fluctuation in the inlet and the outlet are relatively less than in the tooth tip and the cavity of labyrinth seal where have the second frequency spectrum, the third frequency spectrum and the fourth frequency spectrum. The high frequency is caused by the leakage vortex in the cavity of labyrinth seal, so it can be concluded that the vortex frequencies of the tip seal cavity contain both the wheel rotating frequency and the high frequency after breaking up into small scale vortexes through the spectrum analysis.

Analysis of steam exciting force in tip seal clearance

Due to the unsteady effect of the flow field, the pressure fluctuation will happen in the blade tip seal, which will cause the steam exciting vibration force of blade tip clearance changing over time. In order to research the steam exciting force of seal clearance, firstly the pressure field and the velocity field of tip seal clearance are analyzed to calculate the overall distribution of pressure field, and then the steam exciting force to rotor can be presented by solving the integral of pressure on the blade shroud area. Set the pressure on unit area, dA, of shroud surface as P_{r}, integral to the entire surface of shroud, obtain the vibration force of blade tip clearance, and break it down to the y-, z-direction, then the F_y and F_z will be obtained, respectively, as shown in the following formula:

$$F_y = \iint_A P_{\mathrm{r}} \sin\gamma \, \mathrm{d}A \tag{10}$$

$$F_z = \iint_A P_{\mathrm{r}} \cos\gamma \, \mathrm{d}A \tag{11}$$

Figure 8 shows the distribution of the vibration force F_y and F_z under different blade tip seal clearances. It can be seen that the steam exciting force will experience complicated changes over time, and the extent of variation is larger. From the three vibration forces under different tip clearances can be seen, when the blade tip seal clearance increases, the leakage flow can cause greater steam exciting force, this is because the clearance increases, the pressure fluctuation and the pressure of leakage flow greaten, the tangential force and the radial force of shroud will increase accordingly.

Figure 9 shows the frequency spectrum analysis of steam exciting force in the y- and z-direction. The fundamental frequencies of steam exciting force are also 5000 Hz. This suggests that the steam exciting force which is caused by the tip leakage vortex also has the characteristics of periodic fluctuations. The second frequency spectrum, the third frequency spectrum and the fourth frequency spectrum have obvious fluctuations, which due to the unsteady characteristic of complex vortex in blade tip seal clearance, making the vibration force

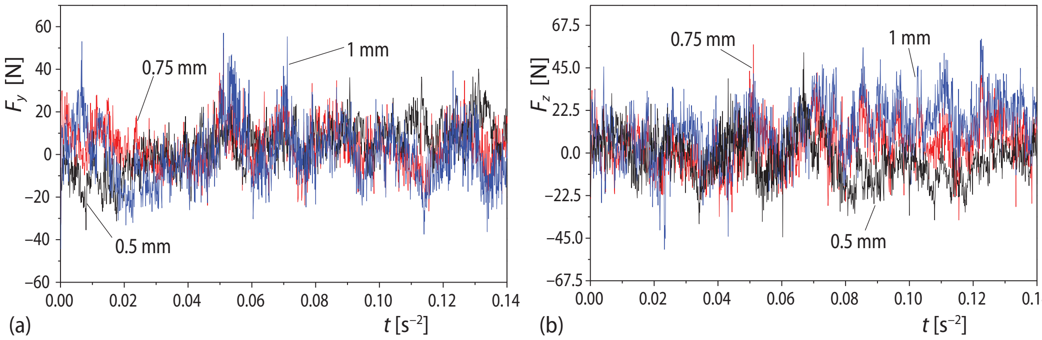

Figure 8. Time analysis of steam exciting force in tip seal clearance: (a) *y*-direction, (b) *z*-direction

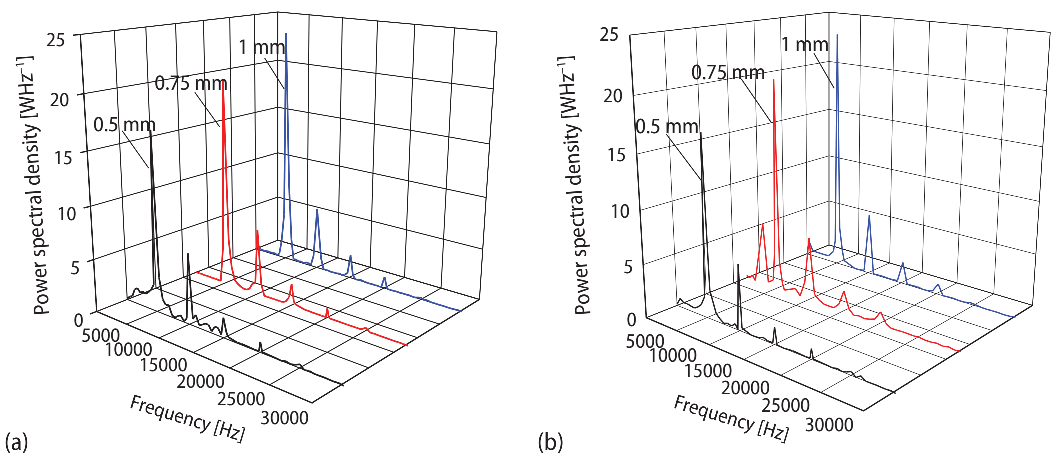

Figure 9. Frequency analysis of steam exciting force in tip seal clearance: (a) *y*-direction, (b) *z*-direction

induced appearing in the high frequency component. The appearance of the high frequency component should be given full attention. The power spectral density, that is the fluctuations of energy, which is corresponded with the fundamental frequency, is big in the larger clearance. Thus, it can be seen that the pressure pulsation in cause of the tip seal vortex increase when the clearance increases.

Conclusions

Based on the numerical investigation of the blade tip seal of steam turbine, the following main observations can be made:

- The stator wake spread and the rotor-stator interaction are the main cause of the uneven circumferential pressure in the inlet of labyrinth seal. The complicated vortex structure of the tip seal leakage flow and the spread of alternately changing stress field between stator and rotor in the labyrinth seal clearance are the main cause of the uneven circumferential stress in the tooth tip and the cavity of labyrinth seal.
- The pressure fluctuations caused by the leakage vortexes in the tooth tip and the cavity of labyrinth seal are complicated. The leakage vortexes are decomposed and collapsed in the tooth tip and the cavity of labyrinth seal, which has an obvious influence on the pres-

sure pulsation at these two positions. The vortex frequencies of the cavity contain both the wheel rotating frequency and the high frequency after breaking up into small scale vortexes through the spectrum analysis.

- The unsteady characteristics of the tip seal leakage flow induce the complicated steam exciting force changing over the time. The pressure pulsation in cause of the tip seal vortex increase with the clearance increasing.

Nomenclature

C_p – pressure pulsation coefficient, [–]
C_s – Smagorinsky constant, [–]
dA – unit area, [m^2]
F – force, [N]
n – rotation speed, [rpm]
p – pressure, [Pa]
$\overline{S}_{ij}$ – rate of stress tensor, [–]
t – time, [s]
U_{tip} – circumferential speed, [ms^{-1}]
u – velocity, [ms^{-1}]
Z – blade number, [–]

Greek symbols

γ – angle between unit vector of the area dA and the co-ordinate axis
Δ – filtering scales, [–]
μ – dynamic viscosity, [Nsm^{-2}]
ν_t – turbulent viscosity, [kgm^{-1}s^{-1}]
ρ – density, [kgm^{-3}]
τ_{kk} – subgrid-scale isotropic, [–]

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