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CALCULATING STUDY OF THE TURBINE AT LAST STAGE FLOW FIELD IN THE SMALL VOLUME FLOW CONDITION

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ABSTRACT

Based on basic equation and boundary layer theory of pneumodynamics, the thesis conducts numerical modeling and theoretical analysis on the last stage of turbine characteristics at a small volume flow by using FLUENT, gives an emphasized analysis on the position of first occurrence of backflow and its expansion direction and comes up with flow structure of the turbine flow field at last stage in the small volume flow condition. In connection with specific experiments, it puts forward the flow model of backflow occurring in the last stage field and the solution to the model. The flow field at last stage for a 100MW turbine in the small volume flow condition that is calculated by using the model is basically in conformity to the actual result.

Keywords: Turbine; Small volume flow; Flow model; Backflow

INTRODUCTION

Running experience of and some searches in modern high-power turbine units show that [1] when the turbine is running in the small volume flow condition, its reliability reduces due to high vibrating stress of moveable vanes at last stage and the exhaust part heated. On the other hand, the last stage of a high-power turbine is generally working in the wet steam zone. Backflow carrying water drops behind movable vanes in the small volume flow condition strikes the movable vanes in a high relative speed, resulting in the water erosion at steam outlet side, which has an influence on the service life of movable vanes and safety of turbine units [2-6]. However, many units often have to run in a very low volume flow, such as idle running and primary load stage at startup, units responsible for auxiliary power when line fault, great extraction of stages behind extraction chamber for heating units, units subject to some peak loads . In addition, some power plants also run under the same condition as the first and second last stage of condensing turbine during increased backpressure heating. In order to ensure high reliability of turbine running and increase its economy, it is of great significance to study on flow characteristics at the last stage in the small volume flow condition [7].

The small volume flow condition is so-called when compared with the design working condition. Small volume flow of turbine means that the condition that the steam flow multiplied by the specific volume is smaller than the specified working condition. In the small volume condition, backflow occurs at root on steam outlet side of movable vanes and eddy forms at the gap between moving and static vanes [8]. Owing to two different flow characteristics, the analysis method widely used in designed working condition is no longer applicable to the small volume flow condition.

As volume flow continuously decreases, steam turbine switches from working condition zone to forced draft zone through transitional condition zone [9]. Where is the switch point? This is also a question about how much the working scope is. Therefore, it is of theoretical and practical significance to study on flow characteristics of last stage flow path in the small volume flow condition.

Flow study of turbine in the small volume flow condition is under progress, mainly focusing on last stages with methods of test research and numerical modeling. Literature [10-15] numerical modeling shows that the flow field in exhaust cylinder consists of passage eddy, separate eddy, end wall eddy, among which passage eddy is the biggest and the major factor affecting the cylinder loss. Literature [16] measures and studies the wet steam flow at last stage low pressure of 300MW direct air cooling turbine using the developed measuring device; Literature [17, 18] conducts numerical modeling analysis of last stage flow field in wet steam zone. Literature [19-21] simulates eddy flow condition of diffusers with different structures in the exhaust cylinder and in different inlets. Satisfactory results have been resulted from present numerical modeling. Compared with test research at high cost, numerical modeling has a great advantage. Current research result has reached consensus on flow separation of last stage vanes in the small volume flow condition.

When mass flow rate G flowing through stage reduces or back pressure increases, volume flow G_v of stage decreases. As volume flow reduces, line flow starts to warp and steam is squeezed to the root in guide vane and towards outer edge in movable vanes. As flow further reduces, this trend is to be intensified. When the volume flow reduces to a certain extent, the flow separation occurs at the root of movable vanes[17]. After flow separation and volume flow decreases more, eddy is formed at outer edge of vane clearance (axial clearance from nozzle outlet edge to movable vane inlet edge). The eddy moves in a peripheral direction in a high speed, which approximates the peripheral speed at the movable vane top in the small flow condition. At this time, the diagonal flow appears in cascade. Volume flow decrease also causes flow redistribution along the vane height direction: flow increases in middle and outer edge parts and decreases in root area. The experiment shows that [7] movable vane root operates in negative degree of reaction under small volume flow direction. Main steam flow only fills outer edge of movable vane passage. Flow at last stages is as follows: main flow to the outlet along vane outer edge, cool and wet steam in vane root condenser side moves from exhaust pipe to flow path. Besides, volume flow decrease and also redistributes of enthalpy drop between guide vane and movable vanes and changes degree of reaction. However, redistribution of both enthalpy drop and flow is caused by fundamental change of flow structures. Things change with a certain rule, so does fluid flow.

MATERIAL AND METHODS

This mathematical model takes basic formula of streamline curvature method [22] as basic equation to describe this flow condition. The difference is to introduce new boundary conditions according to characteristics in the small volume flow condition and make it access to definite answers under new condition.

Flow equation sets of S2 flow plane under a cylindrical coordinate system is derived by using two types of flow plane theory[23]:

$$\begin{aligned}
&C_r\frac{\partial C_r}{\partial r}+C_z\frac{\partial C_r}{\partial z}-\frac{C_u^2}{r}=-\frac{1}{\rho}\frac{\partial p}{\partial r}\\
&C_r\frac{\partial C_z}{\partial r}+C_z\frac{\partial C_z}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial r}\\
&\frac{Di}{Dt}=\frac{1}{\rho}\frac{Dp}{Dt}+T\frac{Ds}{Dt}\\
&\frac{\partial(r\rho C_r)}{r\partial r}+\frac{\partial(\rho C_z)}{\partial z}=0\\
&f(p,\rho,T)=0
\end{aligned}\tag{1}$$

Motion equation is rewritten as the following form for easier solution:

$$\frac{DC_r}{Dt}-\frac{C_u^2}{r}=-\frac{1}{\rho}\frac{\partial p}{\partial r}\tag{2}$$

$$\frac{DC_z}{Dt}=-\frac{1}{\rho}\frac{\partial p}{\partial z}\tag{3}$$

If the flow is broken into flow in planes z and θ as shown in 3-2, component velocity of steam flow speed $\vec{c}$ at any point in z plane is Cu and that in θ plane is Cm:

$$\vec{C}=\vec{i}_u C_u+\vec{i}_m C_m\tag{4}$$

$$\vec{i}_m C_m=\vec{i}_r C_r+\vec{i}_z C_z\tag{5}$$

In which,

$$C_r=C_m\sin\delta\tag{6}$$

$$C_z=C_m\cos\delta\tag{7}$$

$$\frac{1}{r_m}=-\frac{D\delta}{Dm}\tag{8}$$

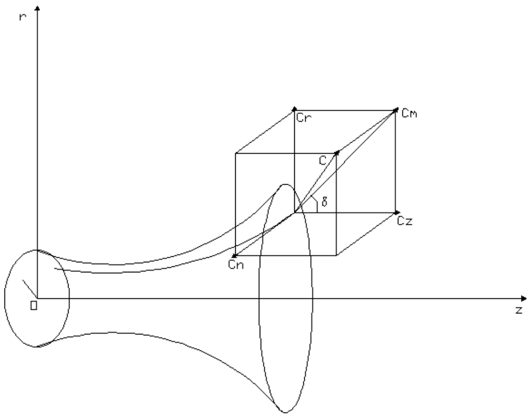

Fig. 1. Flow Velocity Resolution in z Plane and θ Plane.

Substitute Equation (6) and Equation (8) into Equation (2), the full radial equilibrium equation is:

$$\frac{1}{\rho}\frac{\partial p}{\partial r} = \frac{C_u^2}{r} + \frac{C_m^2}{r_m}\cos\delta - C_r\frac{\partial C_m}{\partial m} \tag{9}$$

Quasi radial motion equation under absolute coordinate system is derived from Equation (9):

$$\frac{\partial C_m}{\partial y} = \frac{V(r)}{C_m} - U(r) * C_m \tag{10}$$

in which,

$$V(r) = \sin^2\overline{\alpha}[\frac{\partial H}{\partial y} - T\frac{\partial s}{\partial y}]$$

$$U(r) = \sin^2\overline{\alpha}[\frac{\sin(\delta+\gamma)}{C_m}\frac{\partial C_m}{\partial m} - cty\overline{\alpha}\frac{\partial cty\overline{\alpha}}{\partial y} - \frac{cty^2\overline{\alpha}\cos\alpha}{r} - \frac{\cos(\delta+\gamma)}{r_m}]$$

$$\sin^2\overline{\beta} = \frac{1}{1 + cty^2\alpha\cos^2\delta}$$

Quasi radial motion equation is also derived for relative coordinate system:

$$\frac{\partial C_m}{\partial y} = \frac{V'(r)}{C_m} - U'(r) * C_m \tag{11}$$

in which,

$$V'(r) = \sin^2\overline{\beta}\left[\frac{\partial I}{\partial y} - T\frac{\partial s}{\partial y}\right]$$

$$U'(r) = \sin^2\overline{\beta}\left[\frac{\sin(\delta+\gamma)}{C_m}\frac{\partial C_m}{\partial m} - \frac{\cos(\delta+\gamma)}{r_m} - cty\overline{\beta}\frac{\partial cty\overline{\beta}}{\partial y} - \frac{cty^2\overline{\beta}\cos\gamma}{r} - \frac{2wcty\overline{\beta}}{C_m}\right]$$

$$\sin^2\overline{\beta} = \frac{1}{1 + cty^2\beta\cos\cos^2\delta}$$

$$I = i + \frac{\omega^2}{2} - \frac{u^2}{2}$$

State equation:

$$f(p,\rho,T) = 0 \tag{12}$$

Continuity equation in integral form along y:

$$G = \int_{y_h}^{y_t}\rho C_m\cos(\delta+\gamma)2\pi r dy \tag{13}$$

Energy equation:

$$\frac{D_i}{D_t} = T\frac{Ds}{Dt} \tag{14}$$

Small volume flow condition can be solved as per Equation (10), (11), (12), (13), (14) and corresponding definite conditions.

RESULTS

Flow field at last stage of 100MW single cylinder turbine is calculated using the mathematical model, incorporating actual measurements. Variable working conditions and actual measurements of cross section at last stage movable vanes outlet are shown in Figure 2 and 3, among which ω_1 is calculated relative speed; ω_2 is actual relative speed; c_{z1} is calculated axial speed; c_{z2} is actual axial speed; α_1 is calculated absolute eddy angle; α_2 is actual eddy angle. 60MW load in Figure 2 is non-backflow working condition and 40MW in Figure 3 is the working condition for occurred backflow. Compared with tested parameters and calculated values, deviation between calculation value and test value of non-backflow condition is greater.

Main reason for the deviation is that original data used in calculation is from actual measurement and steam outlet angle has a great influence. In actual measurement, steam flow parameters behind movable vanes are the derived speed and outlet angle from pressure value measured with seven-hole spherical probe. A certain deviation will be generated and it relates to flow field condition, precision of measurement system, adaptability and conversion method. For 40MW with backflow (Figure 3), backflow zone height is 0.13m. From test parameters, height of this zone is within 0.12m-0.15m. This indicates that calculated result basically conforms to actual measurement result.

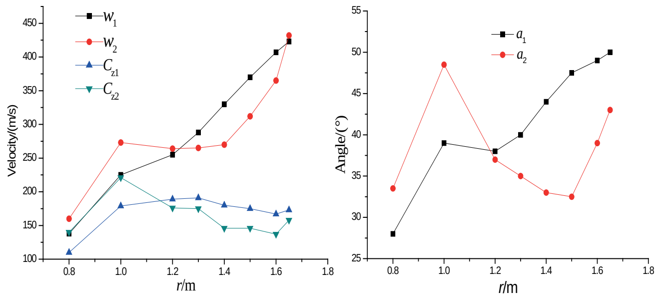

Fig. 2. Contrast between actual measurement and calculation of relative speed, axial speed and absolute eddy at 60MW.

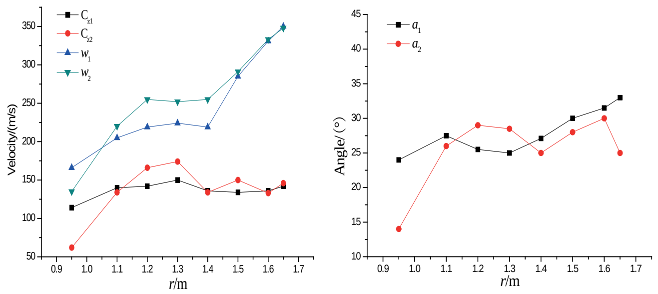

Fig. 3. Contrast between actual measurement and calculation of relative speed, axial speed and absolute eddy at 40MW

CONCLUSIONS

(1) The reason for backflow at last stage in the small volume flow condition: as volume flow decreases, relatively strong diffusion area forms at last stage movable vanes root and pressure gradient forms at outlet cross section.

(2) Simplified mathematical model of last stage flow field in the small volume flow condition is put forward. It basically conforms to actual condition as approximate calculation and reliable to height calculation of backflow area, but approximates and deficiencies exist in the model itself.

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