Steam turbine flow path drainage system under different operating conditions

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Abstract. One of the important elements of anti-erosion protection of the steam turbine last blades is the use of hollow stator blades. There are grooves on the blade surface that lead the steam mixture inside the blades. This mixture is blown to the diffuser area. However, the drain principle fails for low outputs of the turbine, during the so-called ventilation regimes. In the part of blade section the steam flow is reversed and the flow part drain stops working. It was confirmed using experiments of steam turbine carried out in a wide range of power outputs.

1 Introduction

Steam turbine operation in long-term low outputs raises a number of problems. These are caused mainly by a significant output lowering of last stages when the expansion turns into compression. It is accompanied with the occurrence of compression heat and the necessity to cool last rotor blades. Cooling is achieved by injecting water through the nozzles and must be sufficient. At the same time there must be no excessive erosion of the trailing edge of rotor blades near the heavily loaded root section. Furthermore, the compressor operation of the steam turbine blades can cause undesired excessive blade vibrations.

When researching the behaviour of the turbine in low operation, it is always necessary to address the flow and vibration in the area of last rotor blades and the whole stage as a common phenomenon.[1-5]. Many authors are engaged in experimental research of the steam mixture flow near the suction grooves. The dimensions, shapes and locations of the grooves on the surface of stator blades are verified with regard to maximum efficiency of the water film suction with minimal amount of steam sucked from the flow part [6, 7].

The results of experimental and analytical studies are valuable which assume that the water film on the surface of stator blades is not sucked, but on the contrary, that the inside of the hollow blade is heated by warmer steam and the water film evaporates [8, 9]. However, these studies do not address the overall energetic balance of the steam in the turbine. The energy required to evaporate the water film can thus be greater than the energy escaped from the flow part due to the suction of part of the steam.

However, none of the above mentioned works deals with the ability of the drain system to suck effectively the steam mixture direction at low steam turbine output during the so-

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called ventilation. The submitted paper aims to compensate for this shortcoming. The behaviour of this suction tract at nominal and a number of low turbine operations is determined on the basis of a number of experimentally obtained data.

2 Experiment description

Experiments were carried out on the steam turbine of the 200 MW output in the combined cycle power plant. The last stage of the turbine had the rotor root diameter of 1416 mm and the last blade length of 729 mm. Design pressure at the rotor outlet is 11.4 kPa. Root reaction reaches the value of 0.58. Stator blades are hollow. The schematic section of the flow and the last stage with the designation of important elements is in Fig. 1. It is obvious that after the last stage there is the axial diffuser.

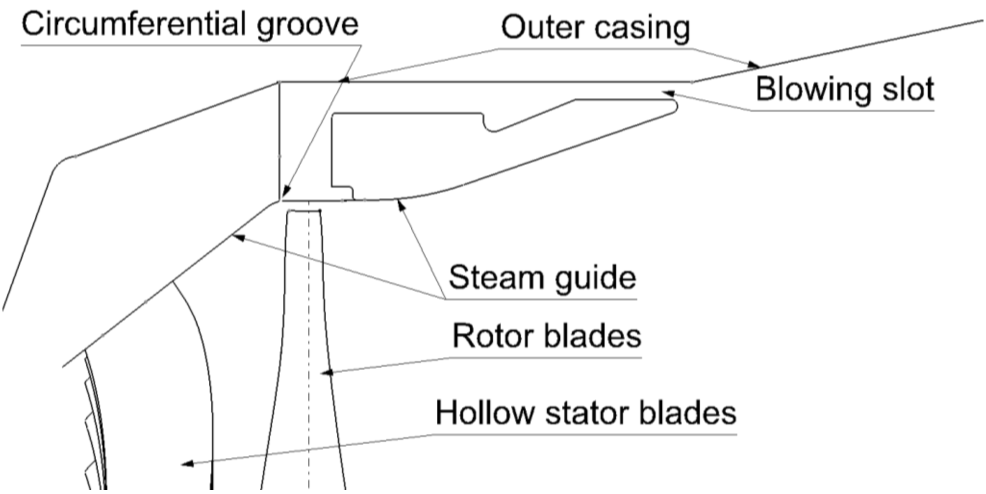

Fig. 1 Steam turbine cross section in the area of last stage tip

In Fig. 2 the points with static pressure samples are labelled using circles. Pressures were measured on the tip limiting wall before (p_{0t}) and behind (p_{1t}) the stator blades, on the tip limiting wall also behind the rotor blades (p_{2t}) and behind the place where the steam sucked from the stator blades is blown back to the main flow (p_{d3t}). The static pressure in the mixing chamber of the diffuser is also measured. All given pressures were measured in pairs at two circumferentially different places of the turbine. Pressure on the blade surface at the suction groove is not measured directly, but is based on a number of CFD calculations:

$$p_{gr} = p_{0t} - 0.45 \cdot (p_{0t} - p_{1t}) \tag{1}$$

For low outputs it is often valid that $p_{0t} \leq p_{1t}$. As the flow in blading increases, all three pressures, namely p_{gr}, p_{0t} and p_{1t}, increase linearly.

Static pressure in the blowing slot was not measured directly. Due to low velocities of the steam in this place (Ma < 0.2) it is supposed that the pressure in this place will be the same as the pressure p_{d3t} that is measured about 100 mm axially behind the blowing slot. Steam mass flow through stator blades is labelled as G_{sb}. The amount of steam G_{hb} is sucked to hollow blades. The volume flow of water through suction grooves is negligible, it makes about 2 % of the total flow through the grooves. Therefore for further calculations it can be assumed that the construction area of the grooves is the same as the area for steam flow rate.

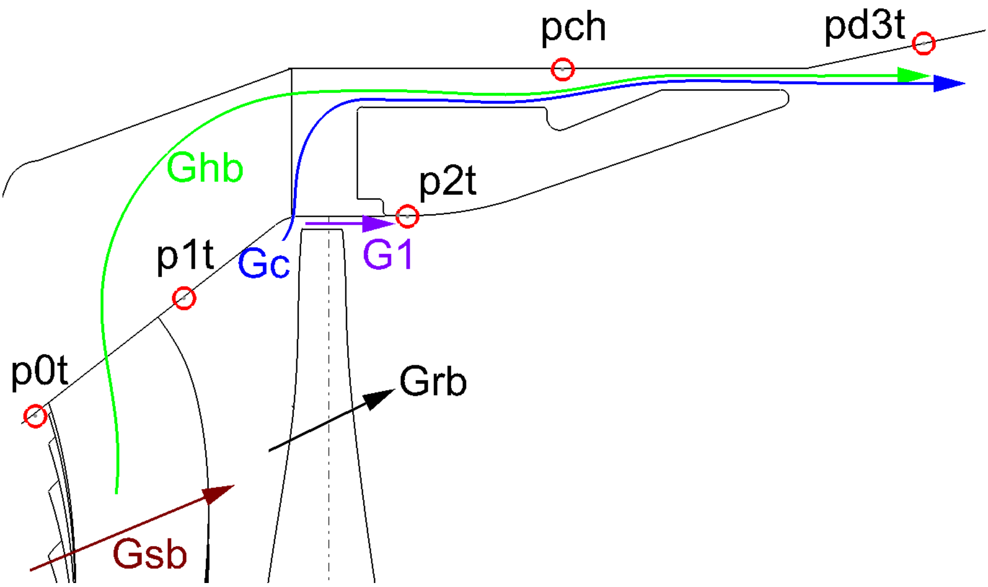

Fig. 2 Measured pressured and calculated mass flows

In equation (2) G_1 is the flow through rotor blades radial clearance and flow G_2 is then:

$$G_2 = G_{hb} + G_c \qquad (2)$$

Here G_c is the steam flow through circumferential groove. It captures a part of water films flowing along the outer circumferential wall of the flow part.
The total steam flow that does not work because it bypasses the last blade is:

$$G = G_1 + G_2 \qquad (3)$$

Analyses were carried out for 8 operating regimes of the turbine differing in turbine output, the flow through the last stage stator blades, outlet axial velocity from the last stage c_{2x} and possibly the so-called ventilation coefficient c_{2x} / u. For values roughly c_{2x} / u $\leq$ 0.17 the turbine is in the ventilation regime, i.e. the regime when the last stage has zero or negative output. The „u“ value is the circumferential velocity at the mean blade diameter. The condenser pressure was in the range of 0.1 to 0.2 bar. The overview of basic parameters for all measured variants is given in Table 1. The measured regimes cover the whole range of the last stage and the exhaust hood work from the zero output to aerodynamic throttling.

Table 1. Overview of measured variants.

Nr.	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
c_{2x} [m/s]	4	40	61	123	209	249	266	360
c_{2x}/u [-]	0.01	0.10	0.15	0.29	0.51	0.61	0.65	0.88
p_{2t} [bar]	0.112	0.164	0.208	0.158	0.121	0.138	0.132	0.100
Output [MW]	0	20	40	122	135	183	198	200
G_{rb} [kg/s]	-	20.5	42.0	63.8	90.3	119.9	125.6	131.0

3 Results

First of all it is necessary to define the pressure ratios over the individual throttling elements of the inner bypass. Specifically these are:

- p_{ch} / p_{gr} (pressure ratio in the diffuser chamber and on the blade surface in the groove area, the pressure loss between the hollow blade and the diffused chamber is neglected),
- p_{ch} / p_{1t} (pressure ratio in the diffuser chamber and on the outer limiting wall between the last stage stator and rotor blades),
- p_{d3t} / p_{ch} (pressure ratio on the outer circumferential wall of the diffuser and the diffuser chamber),
- p_{2t} / p_{1t} (pressure ratio over the last stage rotor blade tip).

For wet steam the critical pressure ratio is 0.583. The decision whether the ratio is critical or not is important for correct calculation of the steam flow. The overview of pressure ratios is in Table 2. Regimes with pressure ratios higher than 1 are marked in red. In these cases there is a back stream flow, in Fig. 2. In this case the suction tract loses its function. Specifically these are regimes 1, 2 and marginally also 3. The output of regime 3 is at the level of 20% of the nominal turbine output. The green fields represent the states when the pressure ratios are critical.

Table 2. Pressure ratios

Nr.	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
p_{ch} / p_{gr}	1.070	1.001	0.935	0.625	0.748	0.376	0.363	0.329
p_{ch} / p_{1t}	1.017	1.033	1.001	0.745	0.868	0.450	0.436	0.394
p_{d3t} / p_{ch}	1.023	1.005	0.992	0.953	0.963	0.848	0.825	0.741
p_{2t} / p_{1t}	1.055	1.063	0.986	0.697	0.834	0.364	0.339	0.266

The relation for critical flow e.g. through the grooves in hollow blades is:

$$G_{hb} = \mu_{hb} \cdot S_{hb} \cdot \chi \cdot \sqrt{\frac{p_{hb}}{v_{hb}}} \qquad (4)$$

It is valid that the coefficient χ = 0.63045, flow coefficient of grooves μ_{hb} = 0.7, flow coefficient of radial clearance leakage μ_{G1} = 0.67, flow coefficient of circumferential slot μ_{Gc} = 0.82, flow coefficient of blowing slot μ_{G2} = 0.96.
In the case of subcritical flow it is possible to calculate the flow (e.g. by blowing to the diffuser wall) as follows:

$$G_2 = \mu_{G2} \cdot S_{G2} \cdot w_{G2} \cdot \rho_{G2} \qquad (5)$$

Density ρ_{G2} is calculated from pressure p_{d3t} and wetness y = 0. Velocity w_{G2} is calculated based on standard Saint-Venant-Wanzel equation, or using relation for critical velocity. The overview of velocities in all parts of the drain system and for all regimes is in Table 3. Negative velocities correspond with places and regimes with pressure ratios higher than 1.

Table 3. Velocities in individual components of the drain system

Nr.	1	2	3	4	5	6	7	8
w_c [m/s]	-77.6	-98.9	-18.4	206.3	291.3	410.6	411.2	411.7
w_{G1} [m/s]	-136.0	-131.5	0.2	233.0	323.7	410.6	411.2	411.7
w_{hb} [m/s]	-136.1	9.5	148.5	294.9	371.3	412.0	412.6	413.1
w_{G2} [m/s]	-41.2	-19.5	25.8	53.7	59.8	114.0	122.9	152.4

Important parameters are relative flow rates, i.e. the values of the flows through the individual parts of the drains related to the flow rate through the stator blade G_{sb}. The summary of flows is in Table 4. Due to low flows through the stator blades and pressure ratios higher than 1, the values of relative flow for regimes 1 and 2 are significantly negative. It is caused by the nature of the flow in the ventilation regime, which is indicated by the static pressure contours in Fig. 3, determined by CFD calculation, for regime 2. Significant compression over the rotor blade tips and higher pressure in the axial outlet area than the pressure at the suction grooves p_{gr} is evident.

Table 4. Relative mass flows in the drain system

Nr.	1	2	3	4	5	6	7	8
G_C / G_{sb} [%]	-7.1	-1.5	-0.2	2.0	0.8	1.6	1.5	1.5
G_{hb} / G_{sb} [%]	-14.9	-0.2	1.4	2.8	1.2	1.7	1.7	1.7
G_1 / G_{sb} [%]	-9.1	-1.5	0.5	1.5	0.6	1.1	1.1	1.1
G_2 / G_{sb} [%]	-22.0	-1.7	1.2	4.8	2.0	3.3	3.3	3.3
G / G_{sb} [%]	-31.1	-3.2	1.6	6.4	2.6	4.4	4.3	4.3

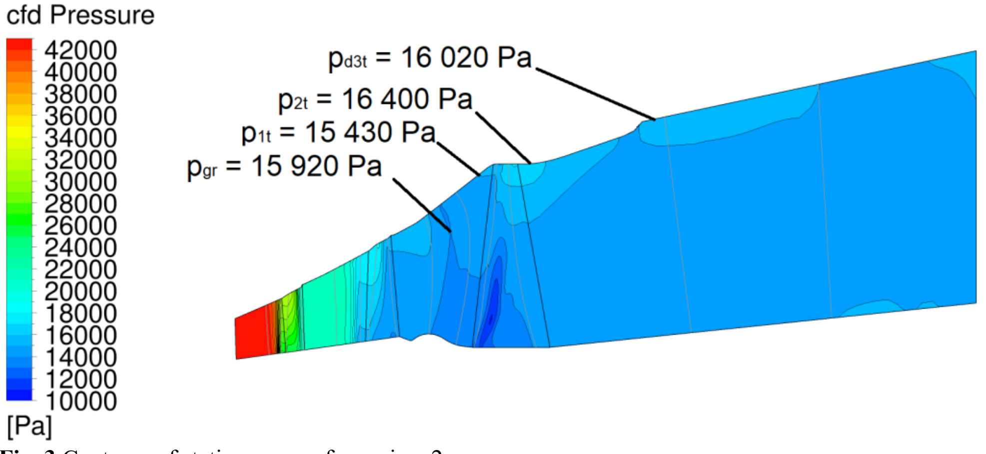

Fig. 3 Contours of static pressure for regime 2

The course of differences of measured pressures p_{gr} - p_{ch} and p_{1t} - p_{ch} depending on c_{2x} / u is shown in Fig. 4. For c_{2x} / u < 0.1 the flow direction in sucking grooves of hollow blades is reversed. However, for the ratio c_{2x} / u < 0.1 the last stage operates in a significant ventilation regime. Long term low output turbine operation is dangerous from the viewpoint of anti-erosion protection of the last rotor blades. In the range 0.1 < c_{2x} / u < 0.17 the turbine

operation is possible, but it is still the area of compression heat generation. It is related with the necessity of cooling rotor blades using water jets and with possible erosion of trailing edges at the root. For values c_{2x} / u > 0.17 it is possible to operate the turbine for a long time without the need for time constraints of operations due to erosion. In the interval 0.17 < c_{2x} / u < 0.6 the turbine is loaded and the flow through the grooves in hollow blades and also through the circumferential groove increases. For c_{2x} / u > 0.6 the drain system is aerodynamically clogged and operates according to design assumptions.
In Fig. 4 also the courses of relative flofws into hollow blades calculated from pressure conditions are added. Sucking from hollow blades works for c_{2x} / u > 0.1, sucking by circumferential groove for c_{2x} / u > 0.16. This value is given by the intersection of the curve $p_{1t} – p_{ch}$ and Δp = 0.

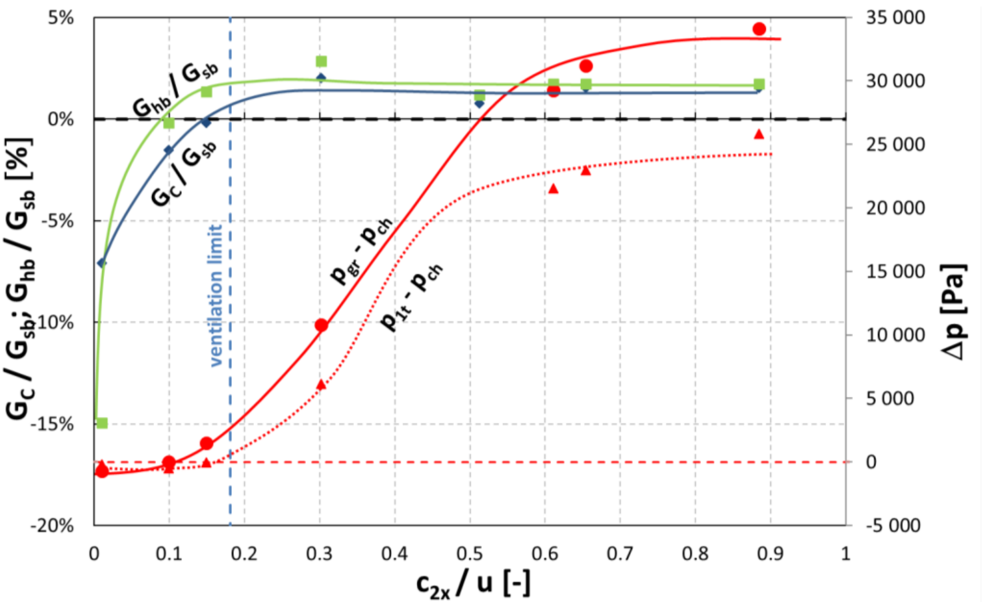

Fig. 4 Courses of pressure differences and relative flows depending on c_{2x} / u for the monitored turbine

4 Conclusions

Extensive experimental research of the last stage flow and the outlet area of the 200 MW steam turbine was performed focused on the functionality of the drain system in a wide range of operations, from zero output to the overload regime.
Based on 10 measured static pressures and geometry knowledge, pressure drops and relative flows in individual parts of the steam inner bypass were defined. For c_{2x} / u < 0.1 the steam flow from the outlet goes back to the stator blades and the drain system does not work. There is a very significant erosion of rotor blades on the trailing edge and when using water jets for outlet cooling even the trailing edge of stator blades can erode at the root. Border limit for long-term turbine operation is c_{2x} / u = 0.17; however, the temperature in the outlet area is also monitored. Now the suction to hollow blades already works effectively and the suction through the circumferential grooves on the tip limiting wall between the stator and rotor blades also starts working.
If long-term operation is required for c_{2x} / u < 0.1, which for the monitored turbine corresponds with 10 % of nominal output, it is necessary to use other elements of anti-erosion protection than just suction using hollow blades. It can be e.g. laser hardening of trailing edges or using hydrophobic surfaces on stator blades.

The work was financially supported by the Technology Agency of the Czech Republic; project FW03010515 Ultra flexible operation regimes of steam turbines and by European Regional Development Fund-Project "LoStr" (No. CZ.02.1.01/0.0/0.0/16_026/0008389).

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