Flow and Vibration in the Small Steam Turbine Last Stage

Michal Hoznedl [1, *], *Tereza* Dadáková[2], and *Jindřich* Bém[1]

[1]Doosan Škoda Power s.r.o., Experimental Research Dpt., 30100, Tylova 1/57, Plzeň, Czech Republic
[2]Doosan Škoda Power s.r.o., Measurement & Diagnostics Dpt., 30100, Tylova 1/57, Plzeň, Czech Republic

Abstract. The paper deals with evaluation of experimental data obtained from the real-size 30 MW output Waste to Energy steam turbine. For many turbine regimes pressures in the last stage area were obtained. At the same time amplitudes of blade tips vibrations were measured of last rotor blades using the blade tip timing system and axial velocity of the steam at the last stage outlet. The obtained data are mutually correlated and relations are analysed between increased vibration values and pressure ratios at the last stage. Increased tip vibrations occur mainly in the area when the pressure ratio over the last blade tip and root is higher than one. In this case there is no expansion, but compression in the last stage of the turbine. The presented results contribute to understanding of processes in the last stage of the steam turbine, mainly for those regimes of turbine operation when its output is lower than 20 % of the nominal output, e.g. during the island regime.

Keywords: *steam turbine, last stage, low load regime, vibration, experiment*

1 Introduction

Due to changes in the energy mix steam turbines often serve as sources of electric energy in industrial applications or in biomass or municipal waste incineration plants: the so called „Waste to Energy – WtE“. It is also valid for the turbine described in this article. Operation of WtE turbines is specific by variable input and output steam parameters. At the turbine inlet mainly steam temperature changes by tens of °C depending on the burnt waste. As a result, at the turbine outlet the steam humidity and, in combination with frequently used air-cooled condensers, also the steam pressure changes. During the day even significant changes of the output occur. The turbine output can, in the long run, be at the level of 15 % or less of the nominal output value.

For the above described reasons the flow of water steam in the last stages and exhaust hoods of small steam turbines is still a current and evolving topic. When designing the last blades, great emphasis is placed on the strength and dynamic design of the last rotor blades. [1-3]. For turbine designers it is important to define well the output, and for low volume conditions even the input of last stages [4-6] using numerical or sometimes even experimental methods. In recent articles [7, 8] aerodynamic instability of the flow has been discussed in the turbine last stages. In this case the flow is solved using demanding DES numerical

* Corresponding author: michal.hoznedl@doosan.com

methods. The paper follows a series of works carried out on first stages of turbo compressors with the rotating stall phenomenon. In [9] the outlet part of the steam turbine is examined for low outputs using experiments and non-stationary numerical calculations.

In the ventilation regime compression heat is generated in the last stage area. In this case under certain circumstances it is necessary to cool the last blade using waterjets located on the outer wall of the axial exhaust hood. The influence of cooling water on the flow at the turbine outlet is examined in detail in [10]. Excessive erosion of leading edges often occurs for ventilation regimes. Due to cooling of steam flow with water jets there is also a risk of erosion of trailing edges. The erosion of the rotor blades is dealt with by authors of article [11].

The turbine blades of last stage which are currently being developed are subject to fundamentally different requirements than the blades developed in the past. Current blades must be mechanically robust and quality anti-erosion protection must be addressed while maintaining a low price and good efficiency. A summary of goals of the current blades development is given in article [12].

However, none of the mentioned articles presents the dynamic behaviour of the blade and aerodynamics of the steam flow on the real (i.e. not experimental) turbine for nominal and low regimes. The submitted paper aims to appropriately supplement the information published so far and discuss the conclusions.

2 Experiment description

2.1 Steam turbine description

Experiments were carried out on a 30 MW nominal output Škoda steam turbine. During the experiments the turbine output was changed in the range of 1 to 30 MW, where the power of 1 MW means island operation of the turbine. In this case the turbine produced energy only for its own waste incinerator. During the measurement, temperature at the turbine inlet and outlet pressure continued to change. The last stage has root diameter 0.928 m, the length of last blade is 0.578 m. Circumferential velocity at the mean blading diameter is 433.7 m/s. The nominal outlet pressure is 6.1 kPa and steam dryness then 0.92.

2.2 Static pressure measurement

Static pressures (index „p“) were measured using wall samples at the root (index „r“) and tip (index „t“) limiting wall. All samples were doubled. It means that pressures were measured at two circumferentially different places on the left (index „l“) and right (index „r“) side of the machine, always15° above the dividing plane. Altogether 12 static pressures were measured before (plane 0) and behind (plane 1) the stator blades and behind rotor blades (plane 2) of the last stage. Average static pressure sample was 2 mm. The reason for the relatively large sample diameter was the fear of clogging the sample with impurities. Impulse pipes were automatically blown every 10 minutes for 5 seconds with atmospheric air. The aim of blowing was to remove possible condensed water steam from pressure pipes. Pressure measurements were done using Netscanner 9022 with pressure sensors Netscanner 9401 in the scope 0-15 psi(a). Combined expanded uncertainty of the measurement for 2σ reached values ±110 Pa at the pressure level of 5 000 Pa. Data were scanned every minute.

2.3 Temperature measurement

In the last stage area altogether ten temperatures were measured at the same positions as for the static pressure samples. For construction reasons only temperatures t_{2rl} and t_{2rr} were not measured, i.e. both temperatures at the root behind the last rotor blade. For temperature measurement flexible K type thermocouples were used with the highest accuracy class. Even for temperatures data were read every minute using cDAQ system from NI. The combined expanded measurement uncertainty was ± 0.4 °C. The last stage scheme including the measurement positions of all pressures and temperatures is shown in Fig. 1.

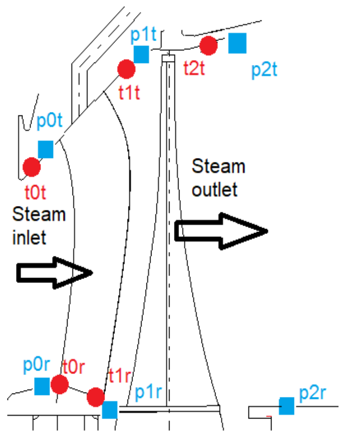


Fig. 1. Last stage.

2.4 Measurement of blade tip timing (BTT)

Measurement of amplitudes of blade tips vibrations was done using in-house developed eddy-current sensors. Data collection was organized using HW from HoodTech Corporation based on NI solutions. Data were processed using special in-house SW that can work with under-sampled data and multiple non-equidistant space sensors. For this measurement altogether 8 sensors were used located in two groups of 3 and 5 sensors. The larger group was placed above the trailing edges and the group with 3 sensors above the leading edges. Two axial positions were chosen for better accuracy of the measurements of vibration amplitudes.

Evaluating in-house SW provides amplitudes of all nodal diameters for each monitored blade. BTT measurement was carried out at the same time as pressure and temperature measurements, that is in a variety of operating regimes of the turbine.

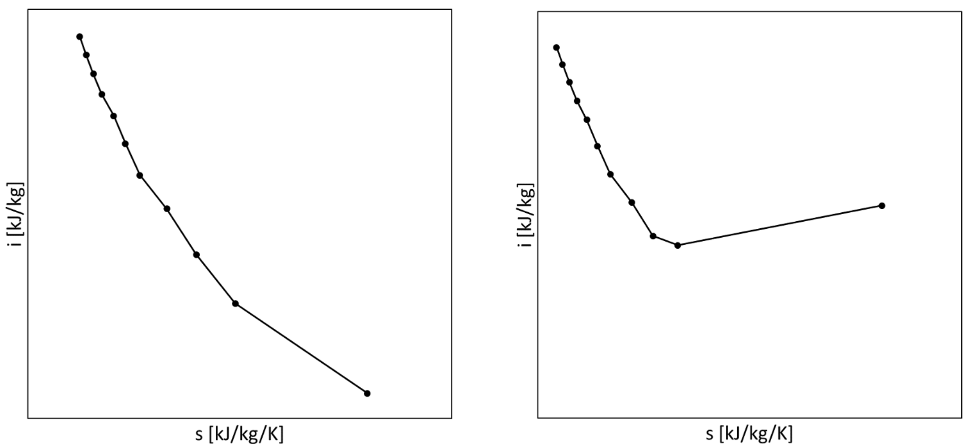
3 Results and discussion

3.1 Nominal and ventilation regime

Experimentally obtained data of vibration amplitudes are evaluated dependent on pressures or temperatures ratio over the root and tip of the rotor blade or the whole stage. At the same time data from BTT are evaluated dependent on the ventilation coefficient φ, when:

$$\varphi = c_{2x} / u \tag{1}$$

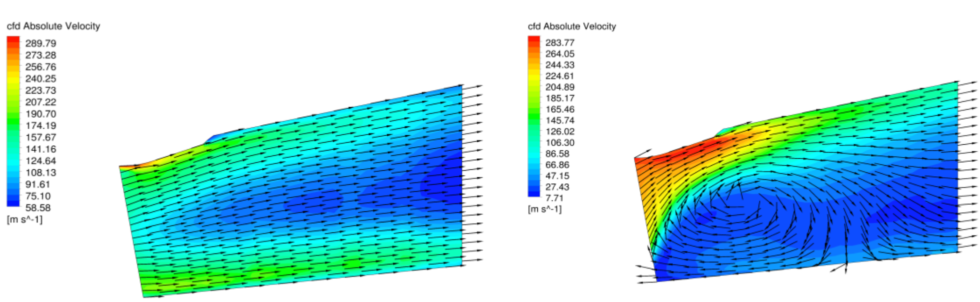
In relation (1) c_{2x} is the axial velocity at the last stage outlet and is calculated from the volume flow of the stage. Quantity u is the circumferential velocity at the mean blading diameter (u = 433.7 m/s). The last turbine stage gets to the ventilation regime when $\varphi_{lim} \cong 0.17$. In this case the output of the last stage is zero. For $\varphi < \varphi_{lim}$ the output of the last stage is negative. The stage, in certain cases even several last stages, consume energy generated by the previous stages. If $\varphi \rightarrow 0$, the machine works with minimal output, it is the island operation. In the ventilation regime of the turbine ventilation coefficient φ must be defined not only for the last stage, but also for the previous stages. However, the value of φ_{lim} is different for each stage. An example of two expansion lines of the last five stages in the nominal and ventilation regime is shown in Fig. 2. For the nominal regime (Fig. 2a) it is valid that $\varphi \cong 0.45$, while for a certain ventilation regime (Fig. 2b) $\varphi \cong 0.15$. In the ventilation regime there is a significant compression at the last rotor blade row accompanied by increased entropy and thus also temperature in the area behind the last stage.



a) **b)**

Fig. 2. Expansion line of last five stages in a) nominal and in b) ventilation regime.

For both nominal and ventilation regime even the flow field in the exhaust hood differs, see Fig. 3. For the nominal regime the exhaust hood is evenly filled with steam flow and velocities at the last stage outlet reach the values about 180 to 250 m/s. In contrast, for the ventilation regime at the tip limiting wall the velocity is high including its circumferential component, while at the root cross-section the steam flow returns to the blading and by the root limiting wall there is a significant separation of the steam flow from the wall. The axial velocity is obviously low.

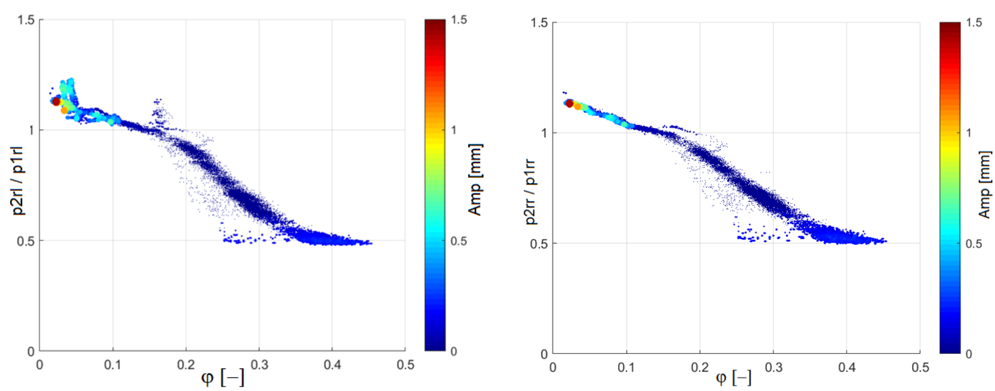


a) **b)**

Fig. 3. Flow field in the exhaust hood for a) nominal and for b) ventilation regime.

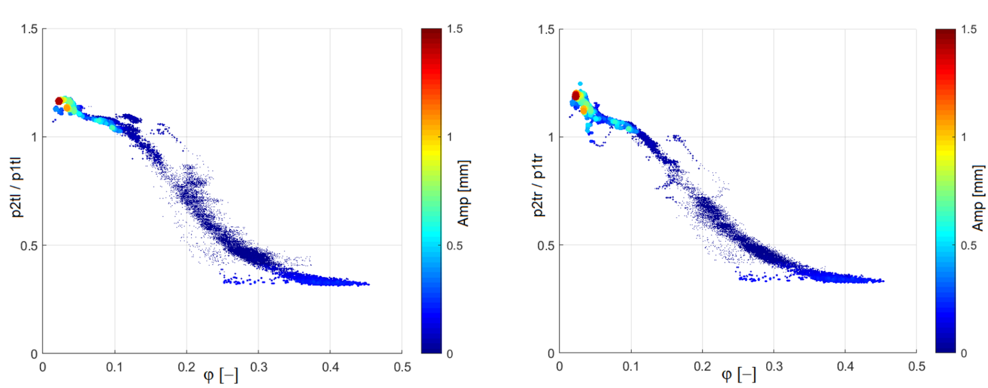
3.2 Flow ratios and vibrations

Pressure ratios at the root of the last stage rotor blade are shown in Fig. 4. Both curves have a similar character. For φ ≥ 0.35 there is no further decrease in the pressure ratio. It stays at value p_{2r} / p_{1r} = 0.5, the blade is aerodynamically clogged even in the root area. Rotor blades vibrations are for φ ≥ 0.35 at the minimum as well as in the interval 0.1 ≤ φ ≤ 0.35. In this interval the blade is gradually loaded, the pressure ratio decreases with increasing φ. Roughly at level φ = 0.15 the negative pressure ratio changes to positive. The steam flow starts to suck through the root cross-section towards the blading. At the same time amplitudes of blade vibrations start to increase. For φ < 0.1 pressure ratio further increases up to the value of 1.1. The side differences in the pressure ratio are beginning to show, given mainly by an uneven steam inflow to pressure samples and by measuring the partial dynamic component of the pressure. For φ = 0.03 significantly higher amplitudes of blade vibrations suddenly occur.



a) **b)**

Fig. 4. Pressure ratio at the last stage root on the a) left and b) right side of the machine.



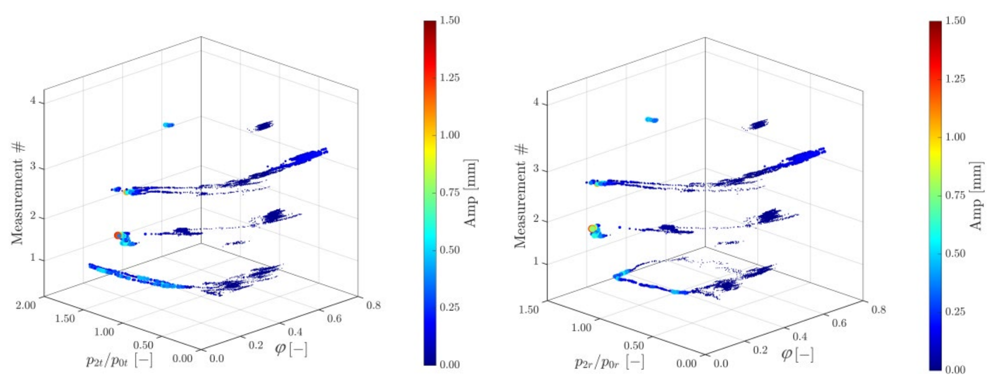
a) **b)**

Fig. 5. Pressure ratio at the last stage tip on the a) left and b) right side of the machine.

Similar behaviour of flow and vibrations as at the root can be observed even for pressure ratios at the tip of the last stage, see Fig. 5. The pressure ratio at the rotor blade tip increased for $\varphi \cong 0.03$ up to the value 1.2. At the maximal value of pressure ratio even a significant increase of vibrations occurs. Value $\varphi = 0$ was not reached, as it would practically mean a state when the turbine is stopped. In the case $\varphi = 0$ it would be valid that $p_2 / p_1 = 1$, thus the pressure in the last stage area (and the whole turbine) is the same.

It should be noted here that in Fig. 4 and Fig. 5 the value of circumferential velocity is related to the mean blading diameter while the pressure ratios are defined at the root and tip limiting section. This is because c_{2x} is a calculated velocity that cannot be defined in individual parts of the outlet cross-section, but only as a mean value over the whole cross-section.

The measurement campaign took place in four periods (four days) numbered below 1 to 4. In Fig. 6 there are graphs that show pressure ratios over the root and the tip of the last stage and the amplitude of vibrations dependent on φ and the measurement period. The course of pressure ratios p_{2r} / p_{0r} in period 1 has a large hysteresis given by the behaviour of the front part of the turbine, namely by gradually increasing and subsequently decreasing of the flow to the regeneration samples of the turbine. Slightly increased vibrations appeared during this period. The pressure in the axial exhaust hood did not change significantly. The most significant increase in vibrations occurred in Measurement #2, when there was a short-term shutdown of the turbine and a subsequent island operation with turbine output of about 0.5 MW for $\varphi = 0.03$. In the next two days no significant anomalies in the character of the measured curves were recorded. The last rotor blade can be excited both by pressure changes at the turbine input and by pressure changes given by the behaviour of the condenser. However, the designed blading can be safely operated throughout the operating range of the machine.



a) **b)**

Fig. 6. Pressure ratio at the tip a) and the root b) of the last stage.

The behaviour of the temperature ratios over the rotor blade tip t_{2t} / t_{1t}, or over the tip of the whole last stage t_{2t} / t_{0t}, is more complicated and is shown for all four measurement periods

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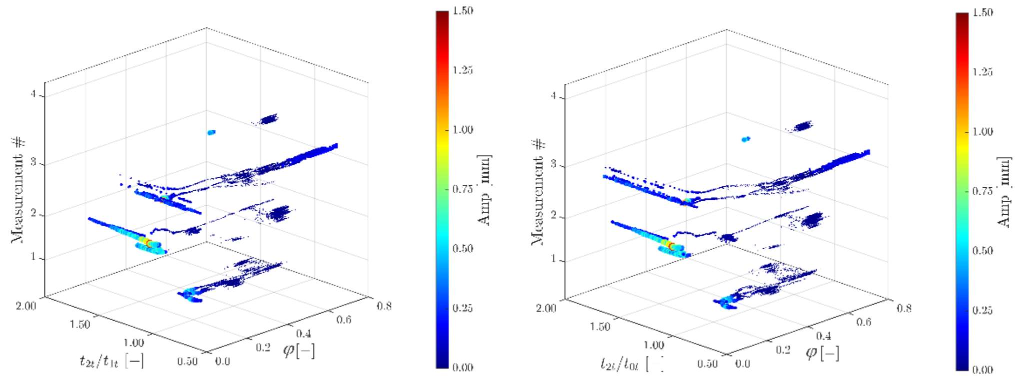
in Fig. 7. Mainly for low outputs or during turbine shutdown the steam temperature behind the stage may increase up to 1.8 times more than the steam temperature before the rotor blade or the whole stage. It is confirmed that the most significant heating occurs in the area of the last rotor blade tip. At the same time, however, the steam heating does not occur in conditions with maximum vibration amplitudes, but especially in conditions that precede maximum vibration amplitudes. In other words, the highest heating occurs when the turbine is in island operation for φ = 0.02 and also at about the level φ = 0.07. Temperature ratios in the tip area are again influenced by both the pressure value in the condenser and the input temperature to the turbine.



a) **b)**

Fig. 7. Temperature ratio over the rotor blade tip a) and over the whole stage b).

The courses of selected values of pressure and temperature ratios together with vibration amplitudes as well as calculated last stage outputs for Measurement #2 are in Fig. 8. The picture also shows the ventilation limit φ_{lim} of the last stage and the ratio of pressures and temperatures that equals one. The pressure in the condenser was changing from 0.04 up to 0.29 bar during the measurement.

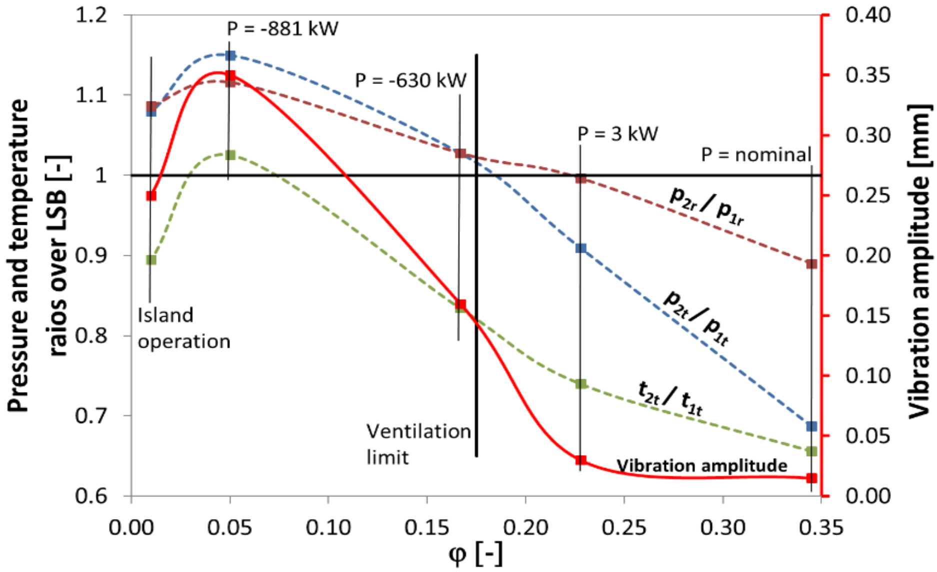


Fig. 8. Courses of pressure and temperature ratios and vibrations in dependence on the ventilation coefficient

When reducing the coefficient φ the pressure ratio over the rotor blade is exceeded first at the root, for the value φ = 0.225. At this point the flow is detached from the rotor blade root. The blade output is approximately zero. At the same time, however, there is an increase in vibrations. For ventilation limit φ_{lim} = 0.18 the steam flow over the blade tip towards the condenser will also stop, ratio $p_{2t} / p_{1t} \cong 1$. At this time the rotor blade already functions as a centrifugal compressor and it has a significant negative output. The value of vibration amplitudes further increases. The maximal value of pressure and temperature ratio and at the same time the maximal value of vibrations and last stage input is for φ = 0.04. At this value of the ventilation coefficient the steam flows significantly radially in the inter-blade rotor channel, centrifuges and dynamically stresses the rotor blade tip. As the flow through the last stage further decreases due to the transition to island operation, the stage is relieved and vibrations are reduced. Pressure ratios are higher than one, the steam is compressed. However, the steam flow is so low that it does not fill the whole inlet cross-section and the steam cools down. It is also possible that the thermometers are not washed at all by the steam flow.

For turbine ventilation regimes it is difficult to measure mean values of pressures and temperatures in the area of turbine outlet at low outputs. There are different values of pressures and temperatures in the root, middle or tip area of the exhaust hood flow profile. When fitting the turbine with operational or research measurement, it is necessary to keep in mind the phenomena associated with the steam flow at low outputs.

4 Conclusions

Four series of experiments were performed on a steam turbine with the output of 30 MW operated in the municipal waste incinerator. At the same time pressures, temperatures and vibration amplitudes of last stage rotor blades were measured. Experiments were carried out in a broad range of the turbine and last stage operation - from the island operation to the overload regime.

For each of four series of measurements the dependences of pressure ratios over the tip and the root of the rotor blade and the whole stage on the turbine operation were determined. Furthermore, the temperature ratios over the tip of the rotor blade and the whole stage for various turbine operations were also determined. Turbine operation means the so-called ventilation coefficient $\varphi = c_{2x} / u$. The limit value is $\varphi_{lim} \cong 0.17$. This is the value for which the last stage output equals zero. For $\varphi < \varphi_{lim}$ the output is already negative and the stage operates in the compressor regime, for $\varphi > \varphi_{lim}$ the stage output is positive. The stage operates in the turbine regime.

A significant increase in the values of vibration amplitudes occurred for the monitored turbine when the steam flow was detached from the rotor blade root, which was indicated by the reversal of the steam flow at the root back to the turbine. When the steam flow was reversed even from the rotor blades tip ($p_{2t} / p_{1t} > 1$) value φ_{lim} was subceeded and the last stage operated in the compressor regime. Maximum values for vibration amplitudes of pressure and temperature ratios were measured for $\varphi = (0.02, 0.07)$. Steam temperature behind the rotor blade tip reached temperatures 1.8 times higher than the temperature before the tip. At the place t_{2t} temperatures up to 210 °C were recorded. For values $\varphi \cong 0.01$ (island operation of the turbine) the steam flow through the turbine was so low that the steam did not fill the whole volume of the exhaust hood and pressure and temperature ratios began to decrease significantly again.

The experiments further verified that the last stage behaviour from the point of view of dynamic loading is influenced both from the front (opening and closing of regeneration samples) and the back (pressure change in the condenser) part of the turbine. When evaluating the measured data it is more appropriate to monitor pressure ratios over the rotor blade than the absolute values of pressures before and behind the blade. The steam flow in the outlet part of the turbine is significantly uneven at low regimes. Mean values of pressures and temperatures can be only determined by suitably selected positions of thermowells and pressure samples at several places of the exhaust hood.

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