Acta Polytechnica CTU Proceedings **20**:43–50, 2018
doi:10.14311/APP.2018.20.0043

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EXPERIMENTAL AND NUMERICAL RESEARCH ON FLOW IN THE LAST STAGE OF 1090 MW STEAM TURBINE

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Abstract. The paper deals with experimental and numerical research in the last stage of real 1090 MW steam turbine with the last steel blade length 1220 mm placed in nuclear power station. The last stage was equipped with twelve static pressure taps. It was also possible to probe in two planes - before and behind the last stage using pneumatic or optical probes. A number of last stage flow parameters were determined at the root and tip wall for nominal turbine output. Among those parameters are static pressures, Mach and Reynolds numbers, last stage reactions and steam wetness. All the directly measured and evaluated flow parameters are taken from locally measured points and that is why even 3D CFD calculation of the whole system - last stage, diffuser and exhaust hood was implemented. Measured and calculated parameters are compared. Especially static pressures are in very good agreement; the only steam wetness has bigger difference due to different measurement position. Locally measured values are enough to estimate the flow behavior of the last stage. On the other hand, the CFD simulations with well determined boundary conditions, meshes and geometry and is effective tool to simulate even very complicated flow structures in the last stage and exhaust hood.

Keywords: Steam turbine, experimental measurement, last stage blade, pneumatic probe, optical probe.

1. Introduction

Flow in the steam turbine last stage and exhaust hood still remains one of the serious problems in the area of steam turbine research. The complexity of the issues can be found mainly in the mutual flow coupling between the last stage, the subsequent diffuser and the exhaust hood. The importance of the research can be easily shown on an example when inappropriate configuration of the last LP stage and the subsequent exhaust hood causes loss up to 24 MW on the turbine with 1090 MW output in a nuclear power plant with six last stages. Specifically for this case the pressure loss between the LSB outlet and condenser inlet is considered at the level of 2000 Pa.

For precise modelling of the whole system of the last stage and the diffuser it is necessary, for numerical simulation, to know exact boundary conditions. Obtaining or deriving exact boundary conditions is often accompanied with significant inaccuracy, sometimes it is even impossible. Certain possibility here can be only the last stage calculation without the subsequent diffuser. However, in this case fundamental differences can be expected between the data obtained by numerical simulation and reality. E.g. in the tip area behind LSB, because of the diffuser effect, velocity will be definitely higher than in the case when the diffuser in the mode is neglected.

For the reasons mentioned above it is recommended to obtain the boundary conditions from experimental measurements. They were in the past carried out on some turbines [1, 2], recently results have been published only on 300 MW turbine with an air-cooling condenser [3, 4]. There are of course many numerical simulations of flow that suitably solve the behaviour of the LSB - diffuser coupling [5, 6]. In these cases boundary conditions are based above all on measurements of experimental steam turbines, e.g. [7].

The aim of presented paper is to show the comparison of experimentally obtained data from the static pressure taps placed on the tip and the root of the 1090 MW steam turbine last stage with the results of numerical CFD simulation. However, in order to obtain other stage parameters (Mach and Reynolds number, stage enthalpy drop) it was necessary to define thermodynamic parameters before and behind the last stage. For this reason traversing was carried out by both the pneumatic and optical probes to get the wetness profile along the blade length. From the measured profiles the points were separated near the root and tip limiting wall and then used for the calculation of the above mentioned quantities. A certain complexity of the problem lies in the fact that the whole last stage is in wet steam, which makes the evaluation of a number of quantities rather difficult. Detailed comparison of pressures and wetness profiles behind L-1 and L-0 rotor blade rows obtained by the pneumatic and optical are out of the range of this paper and they are presented in [8] or in [9].

The second goal of the paper is to compare numerically and experimentally obtained data mainly in the area behind the last stage when firstly the local value of measured and calculated pressure and wetness is compared with the integral value for the entire cross-section. CFD and experimental results must be

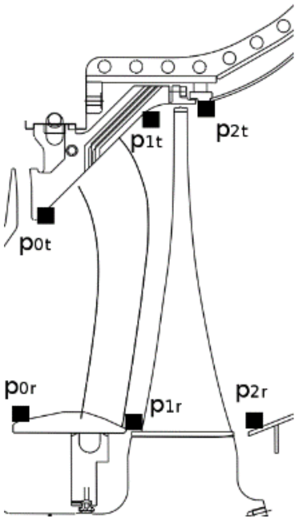

Figure 1. Position of static pressure taps.

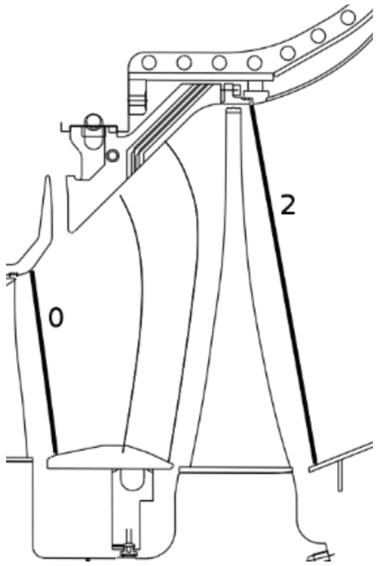

Figure 2. Position of traversing planes.

mutually compared and, if other measured parameters (pressures, velocities etc.) of steam flow along the flow path are consistent, it is possible to consider the CFD simulation validated. Thus the average value can be used to obtain flow parameters in the desired section. An example may be a CFD computation based on the measured boundary conditions before the last stage and the condenser neck outlet and validated using measured data behind the last stage.

2. Experimental setup

One of the six turbine last stages was set up with a number of static pressure taps. Twelve of them were used to explore the flow in the last stage. Static pressure taps were located circumferentially between 9° and 43° above the dividing plane on both sides of the turbine. Meridional position of taps is shown in Fig. 1. It was implemented using stainless steel impulse piping of 12 mm diameter with declination to the turbine. The pressure tap hole had a diameter of 2 mm. On the turbine shroud the taps were connected to pressure sensors NetScanner 9401 with measurement uncertainty of B type on the level of 0.19% from the measured value. The uncertainty value is defined with confidence interval of probability 95.5%. Uncertainties were assessed using GUM methodology. Static pressure taps were equipped with solenoid valves that automatically, at certain intervals connected the turbine space with the surrounding atmosphere. The overpressure forced any water droplets from the impulse piping back to the turbine. It was found after first data evaluation that the pressure p_{1t} has a leakage and it was not used for next evaluation.

In order to define the steam parameters before and behind the last stage the ports for probes with diameter of 50 mm were installed. The probing Plane 0 was in the position of about 70 mm from the last but one stage trailing edges. Its angle to the turbine axis was 80°. Plane 2 was located about 100 mm behind the trailing edges of the LSB stage with the axis declination 85°. Probing was carried out on both sides of the turbine with angle about 45° from the dividing plane. Positions of Plane 0 and 2 are in Fig. 2. Besides both planes also the section was used upstream of LSB, i.e. Plane 1, where it was not possible for technical reasons to carry out probe measurements. Besides probing the steam wetness level using optical probe along the blade length behind the last and last but one stage, measurement was carried out of distribution of pressures, velocities and flow angles [9], as well as probing for defining turbulence parameters and for probing for evaluation of unsteadiness in the steam flow.

Optical probe for measuring steam wetness enables to obtain extinction data that characterize the polydisperse structure of the liquid phase contained in the wet steam.

For wetness evaluation from extinction data more than 40 wavelengths were used in the range 200 - 1000 nm. During measurement the probe was oriented to the steam flow in accord with flow angle α_2, obtained previously from the pneumatic probes measurements. Wetness measurement uncertainty is expected at the level of $\pm 5 \div 10\%$ from the measured wetness value. The whole system was calibrated before measurement using latex particles in the laboratory of Czech Technical University. Picture of the optical probe head is in Fig. 3.

3. CFD setup

CFD flow research methods based on boundary conditions found in experimental steam turbines are a rather common form of flow research in last stages and exhaust hoods [10, 11].

In the given case complex 3D geometry of the system was solved with one last periodically repeated

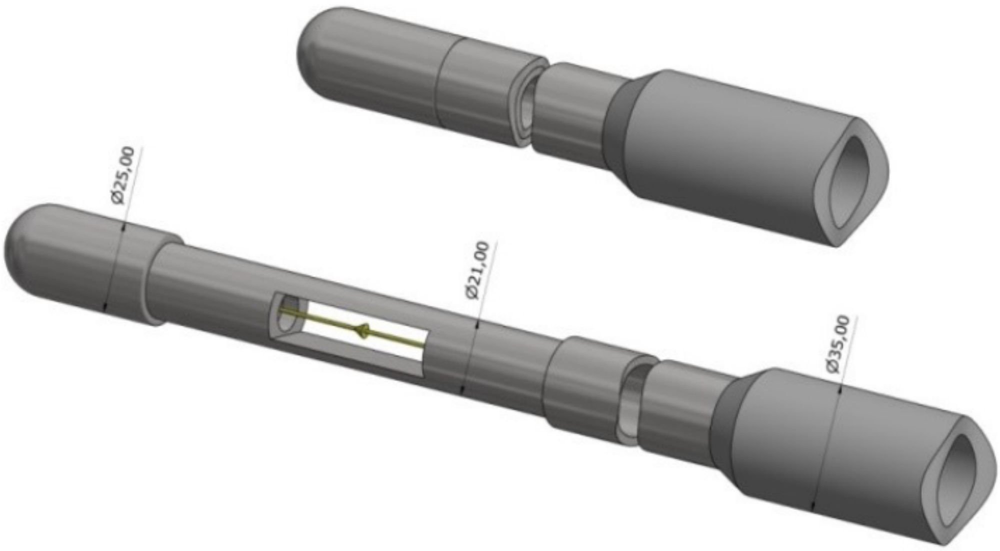

Figure 3. Optical probe head with open and closed measuring space.

rotor blade and all 66 stator blades. The rotor blades outlet is followed by the diffuser, exhaust hood and condenser neck with all important elements, such as the tube reinforcement, low pressure heaters and steam inlet ducts. Only one flow was simulated out of the double-flow exhaust hood. Between the left and the right flow the symmetry boundary condition was entered. For detailed research in the suction slots area even hollow stator blades were modelled as well as the whole suction tract. The last stator blade was modelled including the rotor seal and PSC. These are the elements that significantly influence the quality of transition LSB-diffuser from the viewpoint of flow and its losses.

Between the periodically repeated stator and rotor rows the mixing plane is set due to different spacing of stator and rotor blades. Between the rotor blade and the diffuser, the frozen rotor interface is considered. This setting respects very good distribution regarding the circumferential non-uniformity of the flow entering the diffuser. On the other hand, the disadvantage of the frozen rotor method is the need to model all last blades of the L-0 stage. This makes it more demanding for computational mesh and time.

Simulation was carried out as stationary, fully turbulent (model turbulence SST k-ω) with steam as the working medium, with parameters computed using IAPWS IF-97. For calculations hexahedral computational mesh was used in the blade part of the domain. Hexa-core hybrid mesh was used for exhaust hood and condenser neck meshing. The reason was a large geometrical complexity and irregularity of these components. The whole mesh contained about 60 million cells. The two viewpoints on the solved geometry including the marked boundary conditions are in Fig. 4. The computational mesh is in Fig. 5.

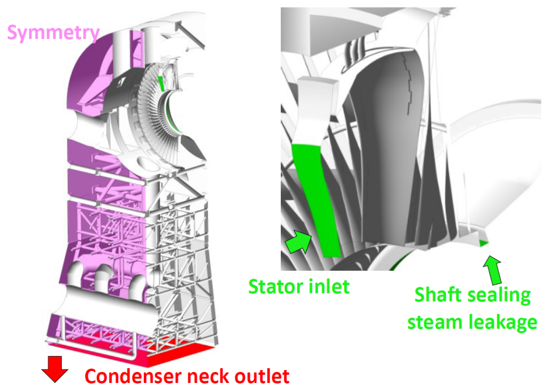

Figure 4. Calculated geometry.

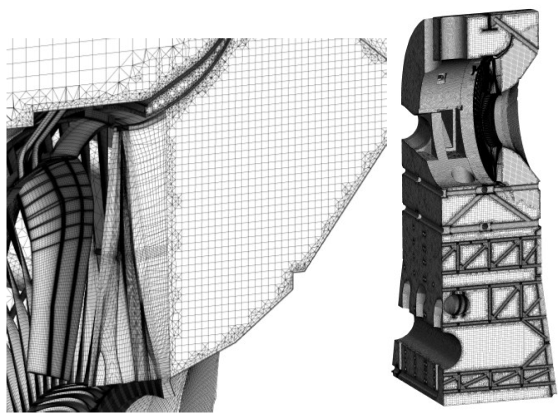

Figure 5. Computational mesh.

As the inlet of the computed domain the profiles of total pressure, temperature, wetness and direction vectors along the L-1 rotor blade length were used. The inlet to the domain is in the position of Plane 0. Data about water steam quantities were obtained by repeated probing. Outlet pressure boundary condition was set with given static pressure obtained by measurement at the same time when the probing was done. After calculation the value of $y+$ parameter was checked. Recommended value of this parameter for the used turbulent model is $y+ < 1$. However, such value was not possible to keep everywhere due to a large computation area. Maximum value of $y+$ was at the level of about 7. Nevertheless, at places where flow phenomena such as flow separation or large pressure gradients can occur the above mentioned condition was met.

4. Discussion of obtained results

A rather significant unevenness of the LSB outlet flow field resulted from CFD calculations. Distribution of absolute Mach number in Plane 2 can be found in Fig. 6. There is an obvious circumferential asymmetry, caused by underloaded last stage and the angle of the outlet steam flow $\alpha_2 > 90°$. Due to this asymmetry a larger amount of flow goes through the right half of the exhaust hood than through the left half. The area behind PSC is also important as due to this obstacle losses are generated there. It influences the local velocity change [10, 12]. Another significant phenomenon is the area of high outlet velocities. These high velocities are generated in places where the loss between LSB and condenser neck is the smallest. It is obviously in the bottom part of the LSB outlet. On the contrary the area of low velocities behind LSB occurs in the upper quadrant of the outlet cross section. The flow coming from these areas must overcome a pressure loss by flowing across the dividing plane and the inlet steam ducts to the LP part. The area of high velocities behind LSB corresponds with low losses areas and vice versa, with the exception of flow in radial slackness. From Fig. 6 it is also obvious that probing (and also static pressure measuring) is in the last stage area in Plane 2 carried out in the places with lower outlet velocity value. Measured values of velocity, pressure and possible other parameters definitely does not correspond with the mean value of velocity in the given area cross section. For this reason it can be unsuitable to implement further data analyses only on the basis of experimental measurement. Thus in this paper basic flow parameters are compared in Plane 0 and Plane 2 and in the stage obtained using CFD as well as measurement. Based on the obtained data it can be judged whether the last stage flow parameters can be calculated only from CFD simulations or whether supported experimental data are needed to confirm CFD at all time.

Static pressure values serve as basic information for other parameters calculation. They are given in Fig. 7, Fig. 8 and Fig. 9. First two columns in graphs always indicate the mean value of static pressure from measurements and from CFD simulations. But these are average pressure values from local pressure taps. E.g. in Plane 0 these are taps p_{0t} and p_{0r}, each on the left and the right side of the machine. The last column shows the mean value of the pressure obtained by averaging the flow in the given cross section.

[%]	EXP	CFD	CFD_integral
p_0		100.3	101.7
p_1	100.0	105.3	100.5
p_2		98.2	91.6

Table 1. Comparison of static pressures in individual planes.

[%]	EXP	CFD
p_{0r}		102.8
p_{0t}		98.1
p_{1r}	100.0	102.1
p_{1t}		107.8
p_{2r}		97.6
p_{2t}		98.9

Table 2. Comparison of static pressures in local points.

A better view of the differences in all three approaches to the pressure evaluation is provided by the data in Tab. 1, when the value 100% always represents the pressure obtained by experimental measurement. It is evident that the biggest differences between individual approaches to evaluation are in Plane 2. The CFD_integral method predicts the pressure value by 8.4% lower than the one obtained by measurement. The reason is circumferential distribution of static pressure that reaches the lowest values in the lower half of the LSB outlet, while the measurement took place in the upper half of the outlet. Significant differences in static pressure are evident also from Fig. 10, where the position of probe traversing is marked.

A detailed view on the above mentioned data is provided in Tab. 2, where local values of static pressures from measurements and from CFD are compared. In Plane 0 and 2 there is a good agreement between experimental results and CFD simulation. A larger difference occurs due to pressure p_{1t}. The cause of the difference is the fact that pressure taps are located just behind the heat extraction that sucks part of the steam outside the flow part of the turbine. This extraction was not considered in CFD. The second cause of the difference was mentioned above - one of the paired pressures p_{1t} was not tight and thus it was excluded from further evaluation.

Wetness value is also an important quantity for defining the thermodynamic state of steam. This value was measured in Plane 0 and Plane 2 using an optical probe. However, for further analyses only two points

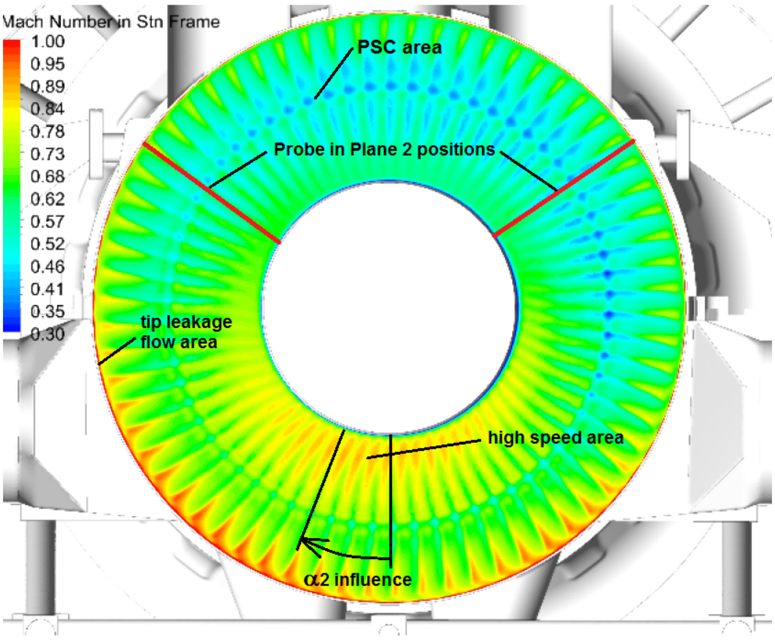

Figure 6. Absolute Mach number in Plane 2.

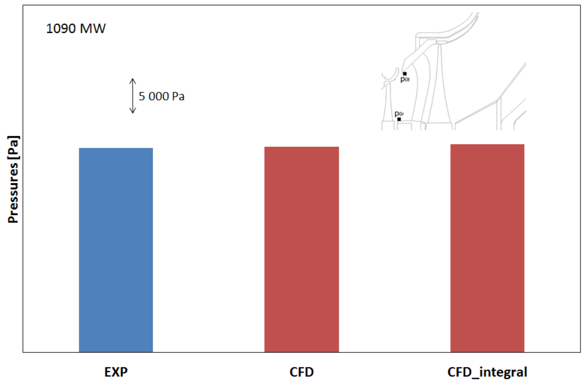

Figure 7. Static pressures in Plane 0.

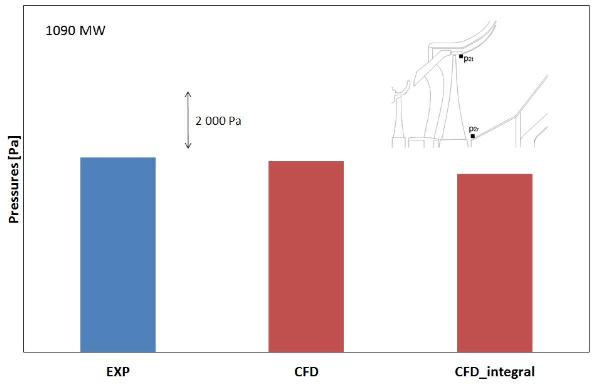

Figure 9. Static pressures in Plane 2.

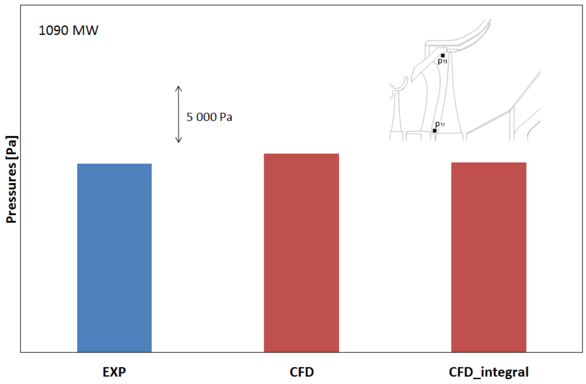

Figure 8. Static pressures behind stator row.

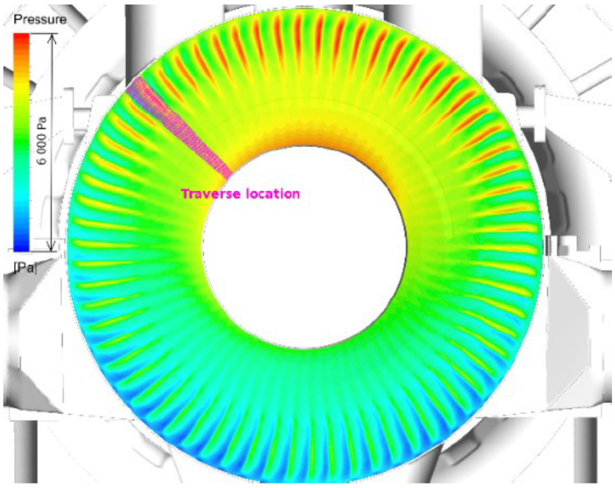

Figure 10. Static pressure flow field in Plane 2.

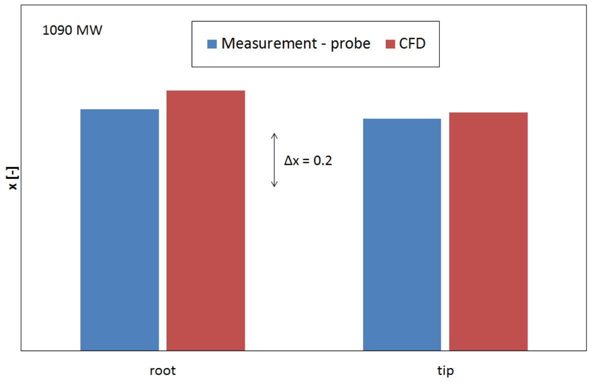

Figure 11. Absolute Mach number in Plane 2.

will be used from the whole profile, one located near the root wall and other one near the tip limiting wall in Plane 2. Comparison of wetness values defined from CFD and from experimental measurement is shown in Fig. 11. The value of steam wetness obtained from CFD is in the root by 7.7% and in the tip by 2.5% lower than the value obtained from experiment. This rather significant difference is caused by the fact that the probe measurement area is located minimally 30 mm above the root limiting wall and is 50 mm long, which is the length of the illuminated space. But in CFD the wetness data are read in points, at the place of static pressure tap. On the other hand, high wetness on limiting walls would mean high local efficiency, which is inconsistent with reality. The reason for this contradiction can be the usage of thermodynamic equilibrium CFD computation of the flow. The results from both CFD and measurement confirmed the fact that higher wetness and thus better efficiency is found near the tip limiting wall.

For further analyses it is important to define individual relations for defining of basic stage parameters. Calculation of thermodynamic parameters that were not measured is derived from water and water steam tables IAPWS_97. Stage parameters of the flow are considered subsequently on the root and tip limiting wall.

The stage reaction is defined based on the ratio of isentropic drop of the rotor to the isentropic drop over the stage in the given cross section, see relation (1).

$$\rho = h_{isN}/h_{isST} \quad [-] \tag{1}$$

Then Mach number of the stage is defined:

$$Ma_{ST} = \sqrt{\left(\frac{2}{\kappa-1}\right)\cdot\left[\left(\frac{p_0}{p_2}\right)^{\frac{\kappa-1}{\kappa}} - 1\right]} \tag{2}$$

Relation (3) defines Mach number for stator blades:

$$Ma_{N} = \sqrt{\left(\frac{2}{\kappa-1}\right)\cdot\left[\left(\frac{p_0}{p_1}\right)^{\frac{\kappa-1}{\kappa}} - 1\right]} \tag{3}$$

In the above mentioned relations it is necessary to put the kappa value $\kappa = 1.11$. This is a common setting for wet steam conditions and it is consistent with evaluation of data from CFD.

Reynolds number of the stage is defined as the last one. The chord of stator blade on the root or tip cross section is used as a characteristic dimension. For Reynolds number calculation it is necessary to define kinetic viscosity ν_2. It is obtained as a result of the function IAPWS_97 for two parameters (p_{2t}, x_{2t}) and (p_{2r}, x_{2r}). Also the isentropic velocity cis from the enthalpy drop over the whole stage is defined:

$$c_{is} = \sqrt{2000 \cdot h_{isST}} \tag{4}$$

In this case h_{is} is defined by function IAPWS_97 from the pressure and dryness in the stator blades inlet and from the pressure and entropy in the LSB outlet. The entropy behind LSB is the same as the entropy in the stator blades inlet subsequently on the root and tip limiting wall. Now it is possible to calculate the Reynolds number of the stage:

$$Re_{ST} = \frac{c_{is}\cdot b}{\nu_2} \tag{5}$$

The values of all described parameters are given or as a local value for root and tip, or in the form of integral value defined only using CFD computations. The integral value is defined from the mass flow weighted parameters of the flow detected over the entire section 1, 2 or 3. Only the integral value for Reynolds number is missing. The reason is that it is impossible to define the characteristic dimension, i.e. the stator blade chord, valid for all its height. Percentage comparison of values is shown in Tab. 3. For evaluation of CFD and of experimental data the same relations are used. Due to considering all the important components and CFD computation of the complex geometry the differences from the experiment results should be minimal. More significant differences can be expected in the stator blade root because of a bigger difference between the measured and calculated wetness value. Further differences can occur on the tip limiting wall due to a higher pressure p_{1t}, obtained from numerical simulation.

The first compared value is the stage reaction. CFD simulations showed a higher stage reaction than the measured one, mainly because of the pressure p_{1t}, that was by 7% higher than that from experiments. On the other hand, the integral value of the reaction is in fact identical with the average value of the reaction from measurement in the root and the tip of blading.

There is a very good agreement for stage Mach number in the root and the tip. The integral value MaST that is not an arithmetic average of the same parameter shows certain difference. It indicates that the MaST values along the stage length will be closer to the values near the tip limiting wall.

There is a difference for Mach number of the stator in the tip (again caused by inaccurate pressure

[%]	EXP	CFD	[%]	EXP	CFD
ρ_r		102.9	Re_{STr}		93.6
ρ_t		109.0	Re_{STt}		96.2
$\rho_{integral}$	100.0	98.7	-	100.0	-
Ma_{STr}		102.2	Ma_{Nr}		100.8
Ma_{STt}		99.9	Ma_{Nt}		90.5
$Ma_{STintegral}$		103.8	$Ma_{Nintegral}$		99.7

Table 3. Comparison of last stage flow parameters.

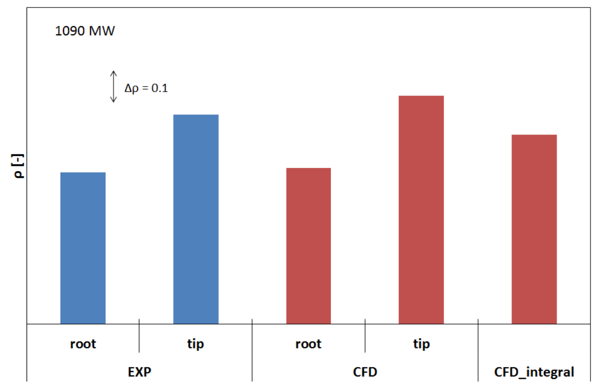

Figure 12. Stage reaction.

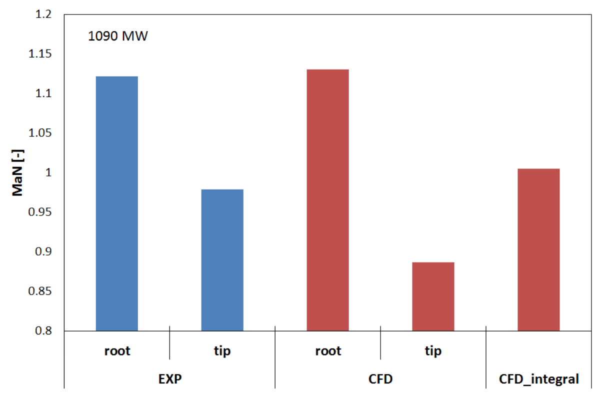

Figure 14. Nozzle Mach number.

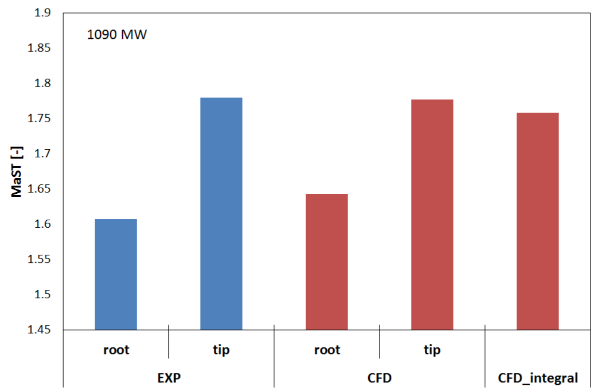

Figure 13. Stage Mach number.

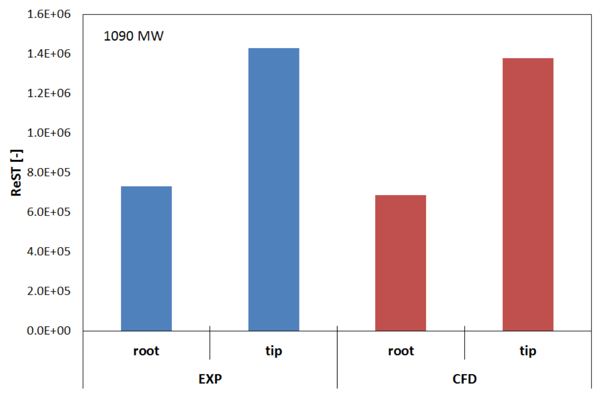

Figure 15. Stage Reynolds number.

p1t). However, the overall agreement is again very satisfactory.

There are also reasonable differences when defining Reynolds number using both methods. Reynolds number is used mainly for calculation of the influence of surface roughness on the stage efficiency. It is therefore not necessary to define it precisely but rather its range.

Static pressure taps in all three planes can be considered as the credible source of information about the entire last stage behaviour. If also the wetness value near the taps is known sufficiently (e.g. from the flow calculation of the stage) it is possible to define many parameters without a demanding probe traversing.

5. Conclusions

Experimental measurement was carried out on the 1090 MW output turbine in the last stage area using static pressure taps. Root and tip steam wetness was defined by probing before and behind the stage. At the same time extensive flow simulations were executed with the same geometry of the last stage, diffuser and exhaust hood as the ones measured on the real work. Due to only negligible differences in the measured and simulated geometries a very good agreement was reached between the results from experimental measurements and CFD computations. It can be said that properly placed static pressure taps can credibly describe the last stage behaviour as for pressures not only in specific points, but also in the entire 2D section. Pressure drops, Mach numbers of the stage and of the stator can be specified accurately. With the knowledge of steam wetness other stage parameters can be defined, such as its reaction, isentropic drop or Reynolds number. Unless detailed knowledge of the flow behaviour along the blade length is required, it is not necessary to execute technically and financially demanding measurement with pneumatic or optical probes, at least not behind LSB.

On the other hand, correctness and accuracy of computational CFD methods was verified using experimental measurement. These methods can accurately predict the behaviour in such a very complicated component of the steam turbine as is the last stage and the subsequent diffuser. However, in order to obtain correct behaviour of computations, it is necessary to accurately specify the inlet and outlet boundary condition. In order to define boundary conditions it is suitable to experimentally determine the flow behaviour at least by using the mentioned static taps. Only this will ensure the results accuracy as for steam flow behaviour also in the places where experimental measurement is very complicated - e.g. between the rotor and stator.

List of symbols

b stator blade chord

c velocity

p pressure

α circumferential angle of the steam

ν kinematics viscosity

ρ density

Abbreviations:

GUM The Guide to the Expression of Uncertainty in Measurement

LSB last Stage Blade

LP low Pressure

PSC part Span Connector

$L-1$ last but one stage

$L-0$ last stage

ST stage

Subscripts:

0 before last nozzle

1 behind last nozzle

2 behind last rotor blades

N nozzle

r root

t tip

Acknowledgements

The work was supported from European Regional Development Fund-Project "LoStr" (No. CZ.02.1.01/0.0/0.0/16_026/0008.389) and from Technology Agency of the Czech Republic from the project TK01020029 Efficiency Increasing of Turbine Wet Steam Last Stages

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