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EXPERIMENTAL RESEARCH ON FLOW IN A 5-STAGE HIGH PRESSURE ROTOR OF 1000 MW STEAM TURBINE

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ABSTRACT

The results of experiments performed on a 5-stage high pressure (HP) steam turbine are presented. Operating parameters close to the real design parameters are simulated. Operating points with slightly different inlet temperature and pressure are also considered. The thermodynamic efficiency based on the torque moment measured by a water brake is compared with efficiency evaluated from the temperatures and pressures. An analysis of velocity ratios and efficiency of individual stages is carried out and the influence of Mach and Reynolds number is investigated. A comparison of the measured and predicted turbine output is also presented.

NOMENCLATURE

b	chord
c	velocity
h	enthalpy drop
l	blade length
n	RPM
p	pressure
t	temperature
u	circumferential velocity
z	number of blades
D	diameter
N	output
Ma	Mach number
Re	Reynolds number
ε	pressure ratio
η	efficiency
κ	isentropic coefficient
ρ	stage reaction

SUBSCRIPTS

0 – 5	measurement planes
b	brake
d	diffuser
h	hub
in	inlet
l	blade
m	mean
max	maximal
mid	middle
n	summation index
opt	optimal
out	outlet
pr	prediction
t	tip
tot	total
tt	from temperatures

INTRODUCTION

The HP part of the 1000 MW turbine is designed with double-flow. In order to perform an experimental verification of the main aerodynamic parameters a 1:1 geometrical scale single-flow variant was manufactured. On the experimental turbine having maximum power of 10MW the course of the expansion through the individual stages was possible to measure and it was possible to determine their efficiency as well as the efficiency of the entire turbine. The experimental turbine enables the inlet parameters of the steam (the pressure and temperature) to be altered, the output pressure cannot be regulated, as it depends on the temperature of cooling water and on the achievable vacuum. On the full-scale turbine at the power station the steam expands into the area of the wet steam in all stages. In the experimental turbine, due to the limited possibilities to regulate the state of the steam, wet steam occurs in two, exceptionally in three stages. In the described work the experiment was focused only on the flow of the overheated steam in all stages to create a

starting database of results for measurement with wet steam. Other aim is to evaluate the impact of deviations of the input parameters of the steam on final efficiency (Hoznedl and Tajč, 2014) and (Hoznedl et al., 2013). Last aim of the measurement is to compare the course of the real and assumed expansion of the steam including the thermodynamic efficiency of the turbine and individual stages. The assumed expansion and efficiency on the base of internally developed Škoda 1D loss model is determined. The loss model is mainly built up on the one-stage turbine, wind tunnel and real power plant measurement. So this is an attempt to improve present loss model by measurement more-stages turbine, especially to try to find the common influence of particular stages each other as well as the cooperation of inlet hood and 1st stage and 5th stage and exhaust hood.

The aim of the experiment is to examine the quality of the proposal and to evaluate the impact of deviations of the input parameters of the steam on final efficiency and prepare a database for wet steam measurement (Hoznedl and Tajč, 2014) and (Hoznedl et al., 2013).

EXPERIMENTAL TURBINE

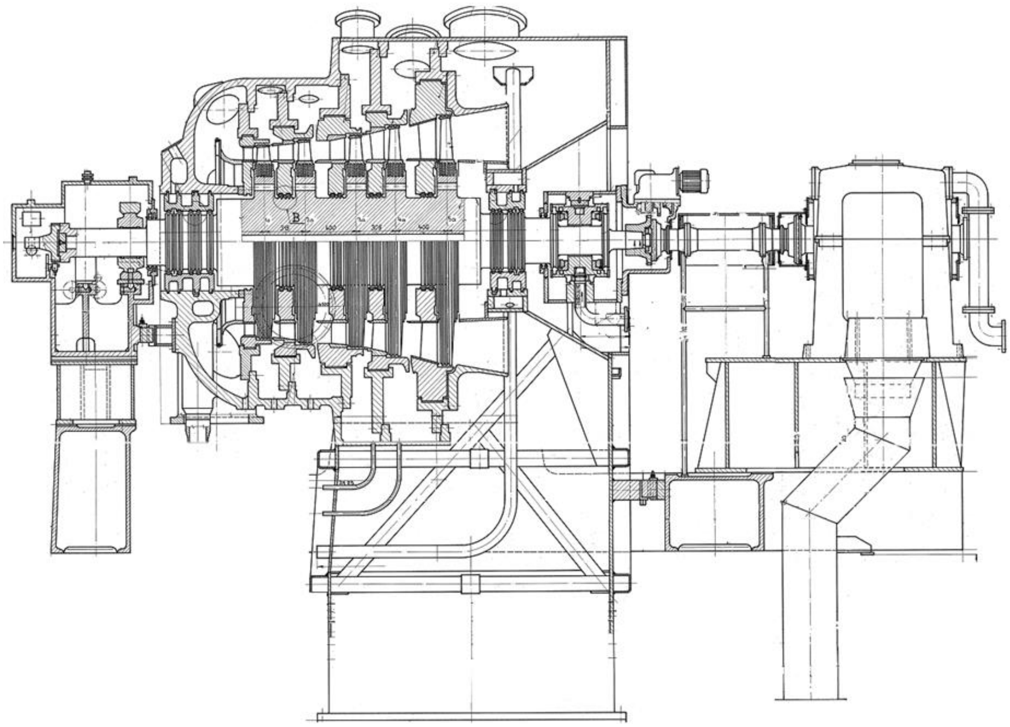

Fig. 1 Experimental turbine

The experimental turbine with a 5-stage rotor is presented in Fig. 1. The temperature between the stages is measured by using resistance thermocouples. Before and behind each stage a pressure tap is installed. Before the 1st stage the total and static pressure is measured by the help of rake probe and behind the 5th stage the pressure on both limiting walls, which means on the hub and tip, is measured. All pressure taps and thermocouples are doubled. They are located on the left and right side of the turbine. There are no sensors between nozzles and blades because of bad accessibility. Maximum inlet temperature is 250 °C and inlet pressure 13 bar. Absolute value of uncertainty of temperature and pressure measurement is ±0.9 °C and ±50 Pa, respectively. Bearings and control system is designed to reach 8000 RPM. The required speeds and the torque moment are regulated and measured by using a water brake. Measured speed has uncertainty at level ± 1 RPM and torque relative uncertainty is lower than 0.15 %. The outer casing includes a number of flanges enabling access to all inner parts of the turbine. These also facilitate the installation of the sensors needed for

measuring the required aerodynamic parameters of the steam at the individual stages. The mass flows of the steam may be specified by the amount of the condensate collected in a measuring tank Maximum steam mass flow is 70 t/hour. Steam mass flow relative uncertainty reached by water tank is ±0.2 %. The input temperature of the steam is regulated by water injection. Turbine is fed by steam from neighbor power plant as well as cooling water. Inlet parameters (temperature and pressure) are kept constant by turbine control system, outlet pressure is dependent on cooling water temperature

The basic data about the blade rows is presented in table 1.

Table 1

	HP 1		HP 2		HP 3		HP 4		HP 5	
	stator	rotor	stator	rotor	stator	rotor	stator	rotor	stator	rotor
b_{mid} [mm]	112.5	115.4	186.8	115.4	185.8	115.4	186.6	118.92	193.8	123.92
l [mm]	99	113	134,7	152	169	187	231	253	309	329.1
D_{hub} [mm]	1258	1250	1258	1250	1258	1250	1258	1250	1258	1250
l/b_{mid} [-]	0.88	0.98	0.72	1.32	0.91	1.62	1.24	2.13	1.59	2.66
z [-]	54	54	36	54	36	54	32	54	38	54
ρ_m [-]	0.14		0.30		0.33		0.38		0.45	

THE CHOICE OF OPERATING PARAMETERS

Operating parameters of the turbine in the experiment were derived from the design data of a real turbine in nuclear power plant. The course of the steam expansion in HP part on the 1000 MW turbine at the power station and on the model in the experimental turbine is shown in the Fig. 2.

Relevant parameters for a real operation were chosen for the experiment in terms of the pressure conditions and the speeds at the individual stages.

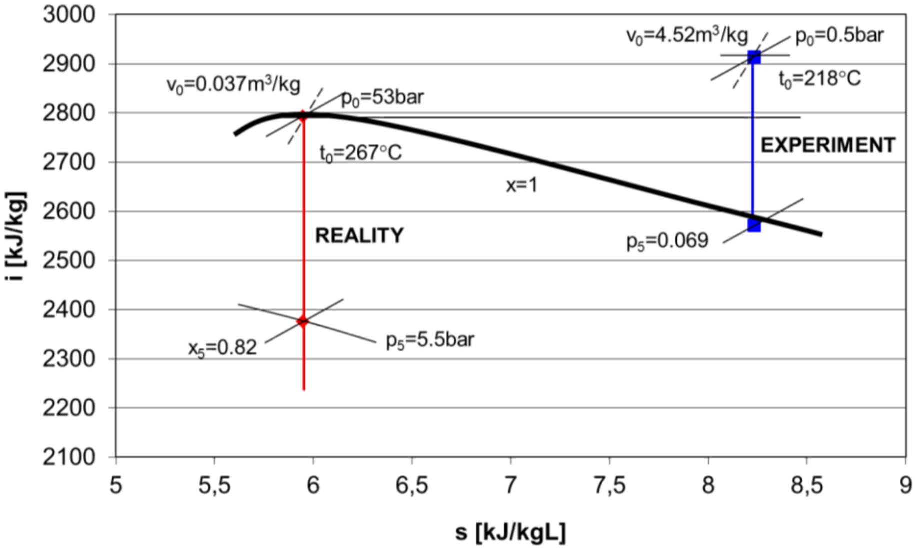

Fig. 2 Expansion on the real turbine and on the model

In the model an effort is made to achieve the same volumetric flow as in the real turbine. However, it is impossible to keep the level of the input enthalpy of the steam. On the model there is the limitation in the output pressure, which considering the quality of the technical equipment and the temperature of cooling water, cannot be lower than 0.069 bar. This also specifies the input pressure and temperature. The recommended operating parameters in the model variant are presented in the table 2.

Table 2

	HP 1	HP 2	HP 3	HP 4	HP 5
p_{in} [bar]	0.5	0.3455	0.2523	0.17	0.1146
t_{in} [deg]	218	180	151	116	84
c_0 [m/s]	400.3	355.8	383.6	367.3	397.8
c_1 [m/s]	388.3	345.1	372.1	356.3	385.9
p_2/p_1 [-]	0.691	0.730	0.674	0.674	0.602
u/c_0 [-]	0.535	0.619	0.5885	0.6428	0.624
c_{ax} [m/s]	77.8	71.7	77.5	74.6	81.1
t_2 [deg]	180	151	116	84	46

n [RPM]	3000
N_{nl} [MW]	2.0488
p_0 [bar]	0.5
p_5 [bar]	0.069
t_0 [deg]	218
t_5 [deg]	46

The scheme of isentropic expansions stated in the Fig. 3 shows how the recommended and performed variants of the experiment differ. Since the assumed pressure behind the turbine was not achieved because of influence of cooling water temperature and no possibility to control it, the higher pressure p_5 = 0.09 bar was used instead of the original pressure p_5 = 0.069 bar, the input parameters of the steam were also shifted to higher values. The variants with higher or lower isentropic gradient were tested too. This differed in the extent of ca 140 kJ/kg. According to the extent of the processed gradient, the individual tested variants are marked from the lowest to the highest by numbers 1, 2 and 3.

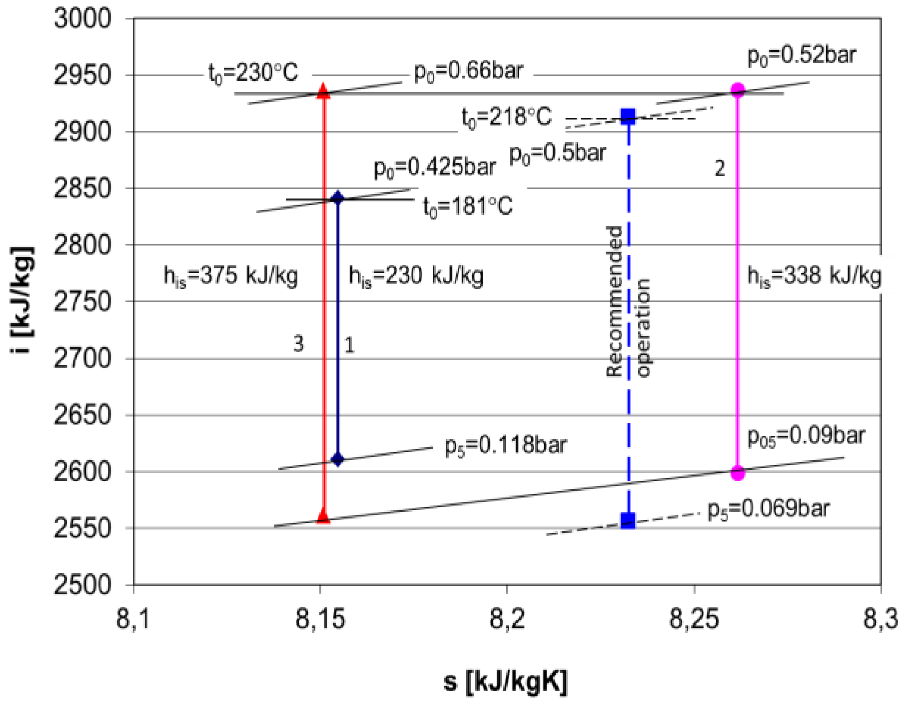

Fig. 3 Summary of tested variants

The different input parameters of the steam influence the mass flow through the turbine. Redistribution of the pressure ratios on individual stages is related to the different enthalpy drops. This also influences the changes in mass flow of each stage.

The efficiency of individual stages depends above all on the velocity ratio u/c, where u is the blade circumferential velocity at the hub diameter and c is the velocity of processed isentropic enthalpy drop. If the stages are well designed, the maximum efficiency agrees with specific rotation speed. For the whole turbine the velocity ratio is considered as the mean value of all stages.

It is valid that:

$$(u/c)_m = \sqrt{\frac{\sum u_i^2}{2 \cdot h_{is}}} \qquad (1)$$

Here h_{is} is isentropic enthalpy

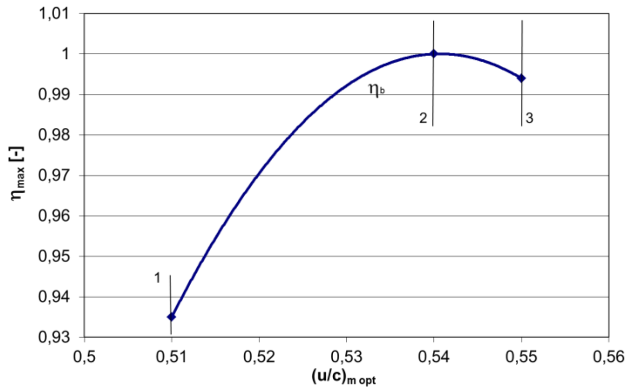

Fig 4. Turbine efficiency

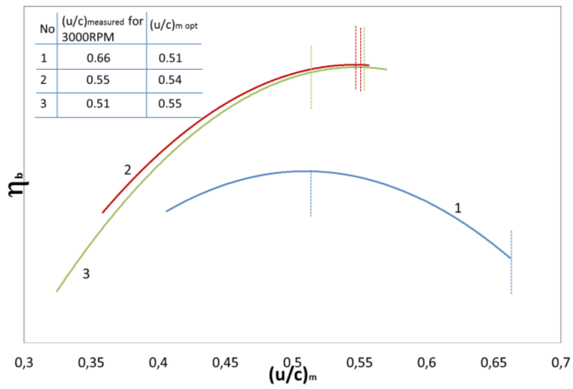

Fig 5. Measured and optimal velocity ratio

The efficiency of the whole turbine can be derived from the torque moment measured at the water brake. The maximum efficiency reached in individual variants is shown in Fig. 4. The best results were reached in variant No. 2, which is close to the experiment recommendation. There are differences in assessment of the optimal velocity ratio for the whole turbine $(u/c)_{m\ opt}$. Fig. 5 shows turbine efficiency for tested alternatives 1, 2 and 3. The value of h_{is} is constant for individual alternatives. Velocity ratio is changed by the help of rotational speed. That is why the deviation of velocity ratio u/c for 3000 RPM from optimal velocity ratio $(u/c)_{m\ opt}$ occurs. The operation of turbine is influenced by cooling water temperature and by condenser pressure. It means that different enthalpy drops are in repeated measurements.

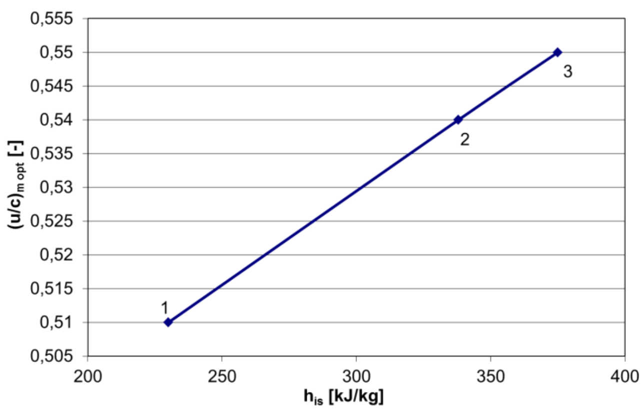

Fig. 6 Optimal velocity ratio

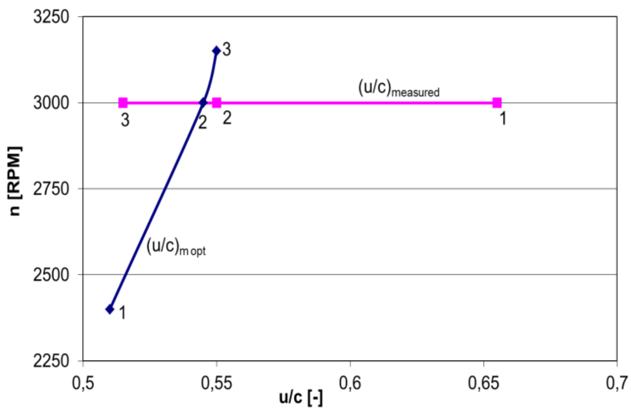

Fig. 7 Velocity ratio at specific RPM

The velocity ratios lead to maximum efficiency are given in Fig. 6. The turbine in the power plant can works at a fixed rotational speed which cannot be adjusted to the immediate enthalpy drop and required output. In Fig. 7 the comparison is shown of the optimal $(u/c)_{opt}$ with measured velocity ratio at specific maximal rotation speed n = 3000 RPM by changing of turbine enthalpy drop, see Fig. 3. The stage efficiency calculated from pressures and temperatures is in Fig. 8 for n = 3000 RPM. The highest efficiency is at 2nd and 3rd stage. The 5th stage is the most sensitive with changes of used enthalpy drop. The efficiency increase by increasing of blade aspect ratio was not

confirmed. There is a negative influence of inlet hood as well. The loss of inlet hood could be up to 4% base on previous CFD calculation. This loss is included to 1st stage efficiency.

Pressure ratios at stages are in Fig. 9 for n = 3000 RPM. Measurement 2 shows the most uniform course of pressure ratios. When the enthalpy drop decreases the pressure loss increases at stages.

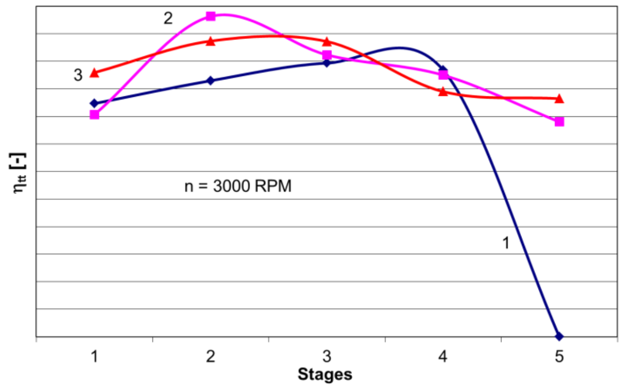

Fig. 8 Stage efficiency

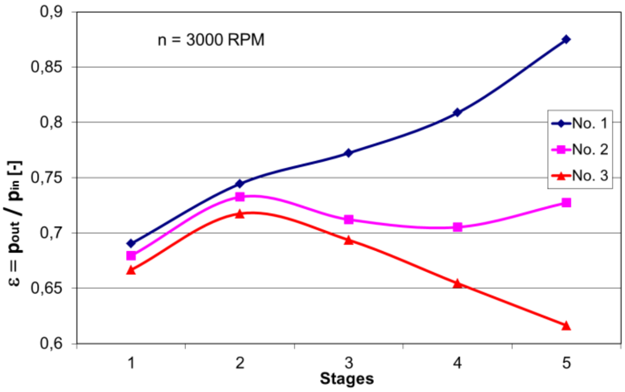

Fig. 9 Pressure ratios at stages

It is obvious that in variant No. 2 good agreement was reached between the operational and optimal velocity ratio u/c. A more detailed analysis of aerodynamic parameters of particular stages will focus on this variant.

MEASUREMENT RESULTS

The distribution of mean values of pressures and temperatures at stages in dependence on the velocity ratio $(u/c)_m$ can be found in Fig. 10 and Fig. 11. It has been always considerably influenced by the cooling water temperature. Contrary to expectation, the designed pressure behind the final stage was not reached. To preserve the pressure ratio over all stages, the inlet pressure had to be adjusted. This corresponds with the shift of real temperatures. Instead of 218 °C the inlet temperature is 230 °C. The inlet pressure increased from the expected 0.5 bar to 0.52 bar before the inlet hood of the turbine.

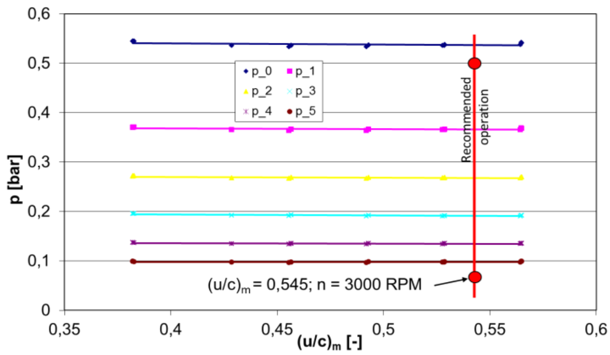

Fig. 10 Distribution of pressures at stages

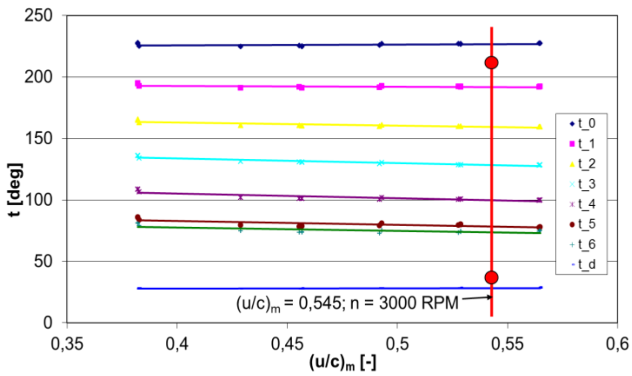

Fig. 11 Distribution of temperatures at stages

Comparison of recommended and assumed and real pressure ratios $\varepsilon = (p_{out}/p_{in})$ at individual stages is found in Fig. 12. For the 1st and 2nd stage the results are comparable. Differences are demonstrated at the other stages, and the largest difference is found in the 5th stage. In fact from the 3rd to the 5th stage at steady inlet pressure a bigger pressure ratio is set up. At the 5th stage it increased from the assumed ε = 0.602 to ε = 0.708 because of different outlet pressure then was supposed. The true is that assumed pressures comes from flow calculation of particular stage and reality means measurement in the gap between stages at the tip wall only, excluding outlet from last stage.

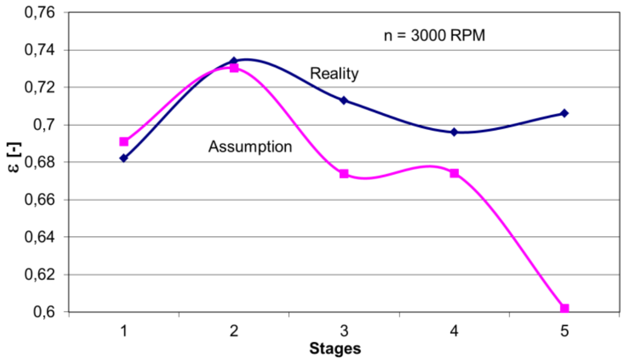

Fig.12 Pressure ratio at stages

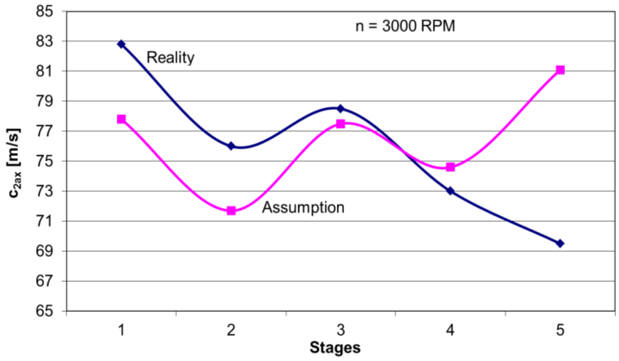

Fig. 13 Axial components of the outlet velocity

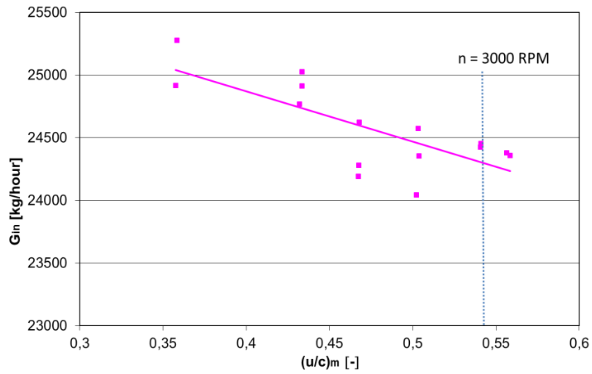

Fig.14 Mass flow through the turbine

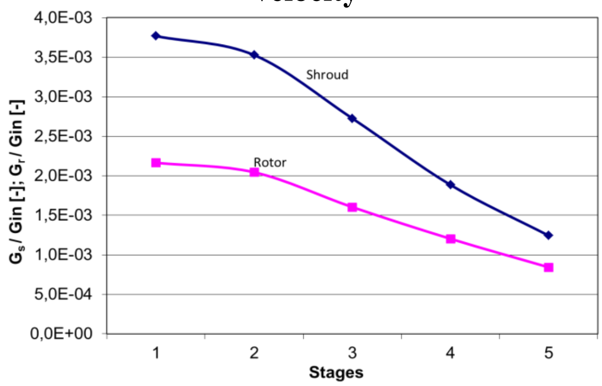

Fig. 15 Steam leak through shroud and rotor seals

Changes of the axial component of the outlet velocity at individual stages are shown in Fig.13. The outlet axial velocity from 1D continuity equation and on the base of measured parameters is determined. Differences appear at all stages. Contrary to expectation, at the 1st and 2nd stage the velocity is higher. The largest difference is found in the 5th stage. For individual mass flow at higher-than-expected pressure and temperature the outlet area at the 5th stage is bigger than necessary. It leads to a reduction of the axial component of the outlet velocity. The 5th stage reacts most sensitively to the deviations from the designed operating states.

With the change of rotation speed the velocity ratios at each stage changes. It is demonstrated by the change of mass flow in the turbine. With the increase of rotation, as confirmed in Fig. 14, the mass flow decreases. At specific RPM, 24.4 t/hour of steam flow through the turbine. The final efficiency is also influenced by steam leaks through shroud and rotor seals. Corresponding mass flows cannot be defined experimentally. As the pressures at the tip and the hub are not measured at the stages, an estimate of the steam flow through the seals is done from the expected stage reaction in the design calculation. The resulting steam flows through the seals are, in comparison with the main flow, very small and always lower than 0.4%. The expected steam leaks at seals are shown in Fig. 15. It can be expected that their influence on the final efficiency will be insignificant.

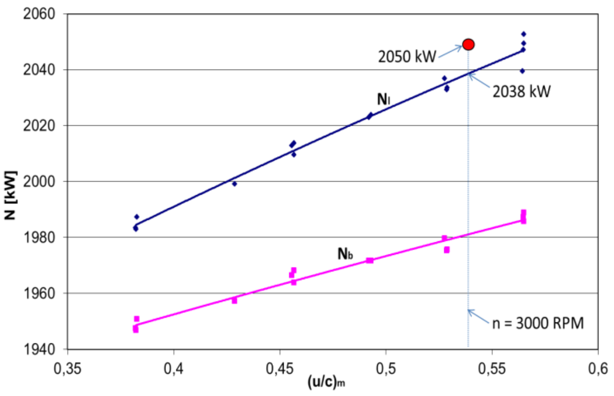

Fig. 16 Output at the brake and the blades of the turbine

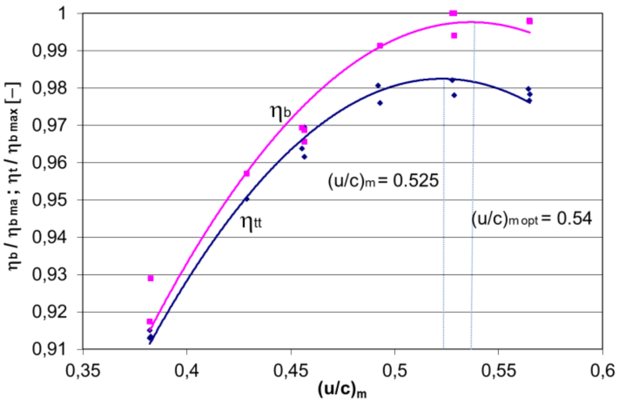

Fig.17 Turbine efficiency

Thermodynamic efficiency of the turbine can be derived from the output given by the torque moment. Output on brake N_b is recorded in Fig. 16. Part of the turbine output is needed for removal of losses in bearings and for friction losses at bladed discs and rotor. Losses in bearings are obtained from measurement. Friction losses are defined by calculation. For rotation speed n = 3000 RPM these additional losses are expressed by ΔN = 58 kW. It is necessary to increase the output on blades by this value. The total output is N = 2038 kW. It is thus 12 kW lower than that expected. It indicates that the final efficiency could be lower than that expected. The difference between expected and measured output can be explained especially by friction loss calculation method which is still empirical one only. There is a demand to test these passive losses in detail during next work.

The thermodynamic efficiency processed using the water brake η_b is shown in Fig. 17. For brake efficiency $(u/c)_{m\ opt}$ = 0.54, which is a value close to $(u/c)_m$ at rotation speed 3000 RPM. The final efficiency is lower than the predicted one. However, in the efficiency prediction the losses caused by Reynolds number and the inlet hood are not included.

The thermodynamic efficiency of individual stages can be estimated only using temperatures measured approximately in the middle of the blade channel before and behind the stage. It is impossible to guarantee mean values for the given flow capacity area. The efficiency assessed in this way has only a referential character. Thermodynamic efficiency of individual stages considering the influence of inlet and outlet flow velocity η_{tt} in dependence on local velocity ratio u/c is given in Fig. 18. It is evident, that the efficiency of the 1st and the 5th stage is significantly worse in comparison with efficiencies of other stages. Conversely, the efficiency of the 2nd stage was extremely high. The steam temperature behind the 1st stage is probably shifted to a higher value. So worsening of the 1st stage efficiency results is improving efficiency of the second stage.

Also the 5th stage efficiency is negatively influenced by the recorded temperature. It results from comparison of efficiencies for the whole rotor, assessed from the brake and temperatures. Efficiency from temperatures is worse by 2.5% for operation at rotational speed n = 3000 rpm. It means that mean temperature behind the final stage should be lower than the measured one. Efficiency of individual stages can be dependent on the states of Mach and Reynolds number.

Stage Mach number for n-th stage is defined on the base of measured pressures as:

$$Ma_{STn} = \sqrt{\left(\frac{2}{\kappa_n - 1}\right)\left(\left(\frac{p_{in\ n}}{p_{out\ n}}\right)^{\frac{\kappa_n - 1}{\kappa_n}} - 1\right)} \qquad (2)$$

where κ comes from steam tables as a function of (p, t) before n-th stage.

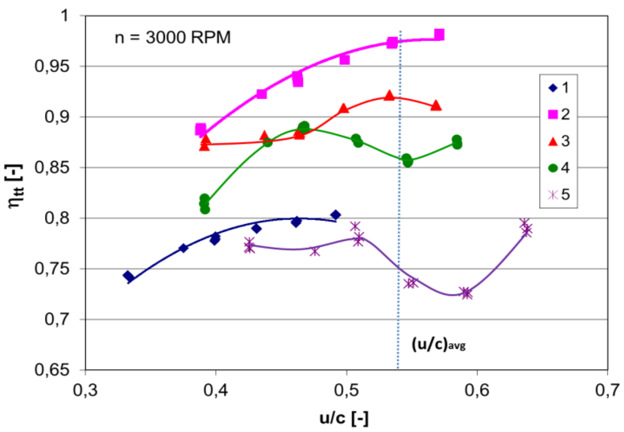

Fig. 18 Efficiency at individual stages

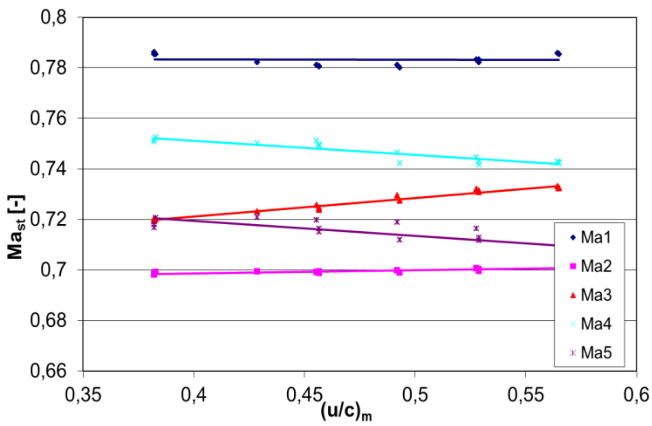

Fig. 19 Ma number course in stages

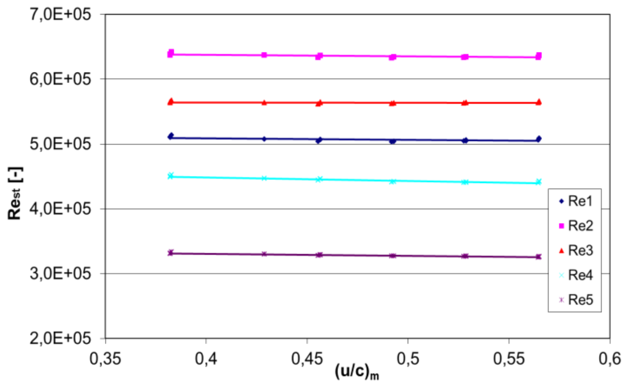

Fig. 20 Re number course in stages

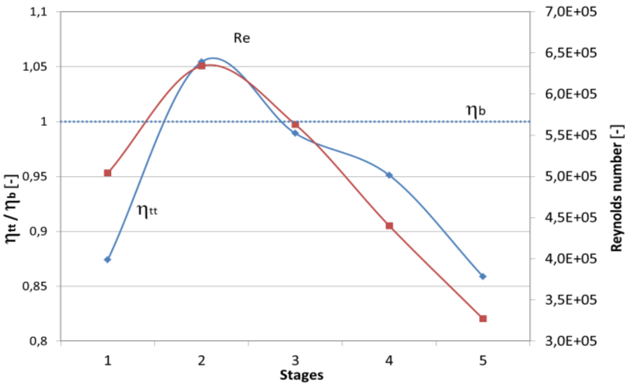

Fig. 21 Efficiencies at individual stages

Ma number courses for all stages are on Fig. 19. The Mach number in all stages is subsonic. Thus more distinctive influence and relation with the final efficiency cannot be expected because all used profiles are designed as subsonic.

Stage Reynolds number is defined on the base of measured pressures and temperatures as:

$$Re_{STn} = \frac{c_{is\,n} \cdot b_{nozzle\,n}}{v_n};\; c_{is\,n} = \sqrt{2000 \cdot h_{is\,n}} \qquad (3)$$

where ν comes from steam tables as a function of pressure and isentropic temperature behind n-th stage.

Certain influence must be considered caused by friction forces that influence the value of Reynolds number. Reynolds number courses are shown in Fig. 20. At given roughness of blade surface Ra ≈ 0.8 the area of transition into the area with independence of the Re number from losses is up to the border $Re > 1.6 \cdot 10^6$. (Tajč et al. 2006). It is evident that in this case it is necessary to assess the impact of friction on losses in all stages. This is confirmed also by the findings from experiments with increased roughness of the blade surface (Tajč, 2005) and (Benetka et al., 2006).

Efficiency of individual stages derived from measured temperatures and pressures is shown in Fig.21. The worst efficiency occurs in the 1st stage and the best in the 2nd stage. In the 1st stage besides the negative influence of temperature the loss in the inlet hood applies as well because temperature t_{in} is measured in inlet chamber of turbine not directly before the 1st stage. In the 5th stage the dominant influence of friction applies that causes a drop of profile losses.

In all stages the efficiency is probably influenced by increased friction. It is also indicated in comparison of the course of Re number to efficiency, which is shown in Fig. 21. Here also the efficiency for the whole rotor η_b can be found. From this comparison it is obvious that assessing efficiency using measured temperatures must be approached very cautiously.

CONCLUSIONS

The experimental work allowed obtaining of aerodynamical parameters for the whole turbine and particular stages as well. Achieved results make possible to evaluate influence of wet steam in particular stages by the change of enthalpy drop processed.

The experiment proved the maximum efficiency close to the designed operating parameters. In other operating points the velocity ratio u/c is not optimal in compliance with the rated speeds of the turbine.

On the experimental turbine the expansion differs from that from the real turbine, only in an area of superheated steam. Hence it is impossible to define the loss due to the influence of the wet steam.

Efficiency of the turbine is evaluated by using a water brake. Efficiency of the individual stages may be evaluated only by using the measured temperatures and pressures. Efficiency of the turbine processed by using the temperatures is different from efficiency evaluated from the torque moment on the brake. Optimal values of the velocity ratio differ too.

The last stage of the turbine reacts to the changes of the processed enthalpy gradient the most sensitively.

The final efficiency is influenced by operation of the experimental turbine out of the area where Reynolds number does not influences losses.

There are particular slight uncertainties in efficiency at the individual stages. It mainly concerns 1st stage.

Mach number at the stages is subcritical, and the flow is subsonic. The influence of Mach number on efficiency is unimportant.

ACKNOWLEDGEMENTS

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