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STEAM FLOW THROUGH A TWO-STAGE TURBINE WITH ROTOR DRUM ARRANGEMENT

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ABSTRACT

The findings of the experimental research of the two-stage turbine with the rotor drum arrangement are presented. The thermodynamic efficiency of both stages established by means of the torque measured by a torquemeter as well as the efficiency evaluated from the temperatures is considered. The influence of the Reynolds number on measured parameters is checked. The shaft steam sealing is modelled and the effect of the balancing slots covering on the resultant efficiency is evaluated.

NOMENCLATURE

b	stator blade chord
c	velocity, total state
G	mass flow
h, i	enthalpy drop, enthalpy
Ma	Mach number
Mk	torque
N	output
p	absolute pressure
P	output
Re	Reynolds number
Ro	stage reaction
s	entropy
T	temperature
u	circumferential velocity
η	efficiency
ω	angular velocity
ρ	reaction
ν	kinematic viscosity
κ	isentropic coefficient

Subscripts

0, 1, 2, 3, 4	measuring sections
avg	average
b	bearing
h	hub
is	isentropic
sb	stator blades
rb	rotor blades
rv	over-shroud seal
s	seal
ST	stage
t	temperature, tip
tq	torquemeter
ur	shaft seals
v	windage
z	axial direction

INTRODUCTION

The rotor drum arrangement enables to extend the lengths of the blades to be installed on the turbine with the impulse stages. This way the decrease of the proportion of the secondary loss is achieved. The disadvantage is the lack of space to be able to install the balancing holes under the stage of the blade section. The slots being the part of each blade are brought into use in practice. A set of small slots, whose number corresponds to the number of blades, is used instead of several larger orifices. The steam from the shaft seal gets directly to the slots throughout the circumference of each wheel. The differences of the pressures on the slot as well as the flow angle to these slots determine the amount of the steam flown through the slots. The number of the measurements at the stages of the experimental turbine with very short blades was carried out whereas the effect of the flow parameters on the efficiency was tested Hoznedl et al. (2009, 2010, 2011) and (Benetka and Valenta, 2003). The aim of the experiments described in this study is to specify the pressure ratios not only on the blades and the over-shroud and shaft seals, but also on the balancing slots. There is an effort to estimate the impact of the steam leakage through the slots and the seals on the thermodynamic efficiency of the individual stages as well as on the whole two-stage turbine.

DESIGN OF TURBINE STAGES

The design of the turbine flow section is illustrated in Figure 1. The chart shows the pressure and the temperature measuring the points and the marking of the local mass flows. The input steam temperature T_0 and the output temperature T_4 are measured by means of the resistance thermometers located in the input and output turbine chambers, i.e. in the areas with much lower speed. The temperatures behind the individual stages (the temperature T_2 and T_3) are sensed by means of the thermocouples. The basic data on blading the individual stages are stated in Table 1. During the experiments the clearance in the over-shroud seals was s = 0.22 mm. The flow through the shaft seal at the first stage was maintained at values close to zero. The input steam parameters are stabilized at the values T_0 = 150°C and p_{0avg} = 0.873 bar due to the turbine control system. The pressure behind the second stage was maintained by means of the control system at values p_{2avg} = 0.405 bar. The shaft torque is measured by a torquemeter and by a water brake. The angular velocity was changed in order to change the velocity ratio in the range from 2100 rpm to 4000 rpm. Uncertainty of the efficiency measured by the torque is ± 0.53%.

Cascade	Stage 1		Stage 2	
	Stator blade	Rotor blade	Stator blade	Rotor blade
Chord b [mm]	27	19.8718	27	20.1735
Length l [mm]	22.7	23.2	25.3	27.8
Number of blades z [-]	172	228	170	258
Aspect ratio l/b [-]	0.767	1.1675	0.937	1.378

Table 1 Turbine stage characteristic

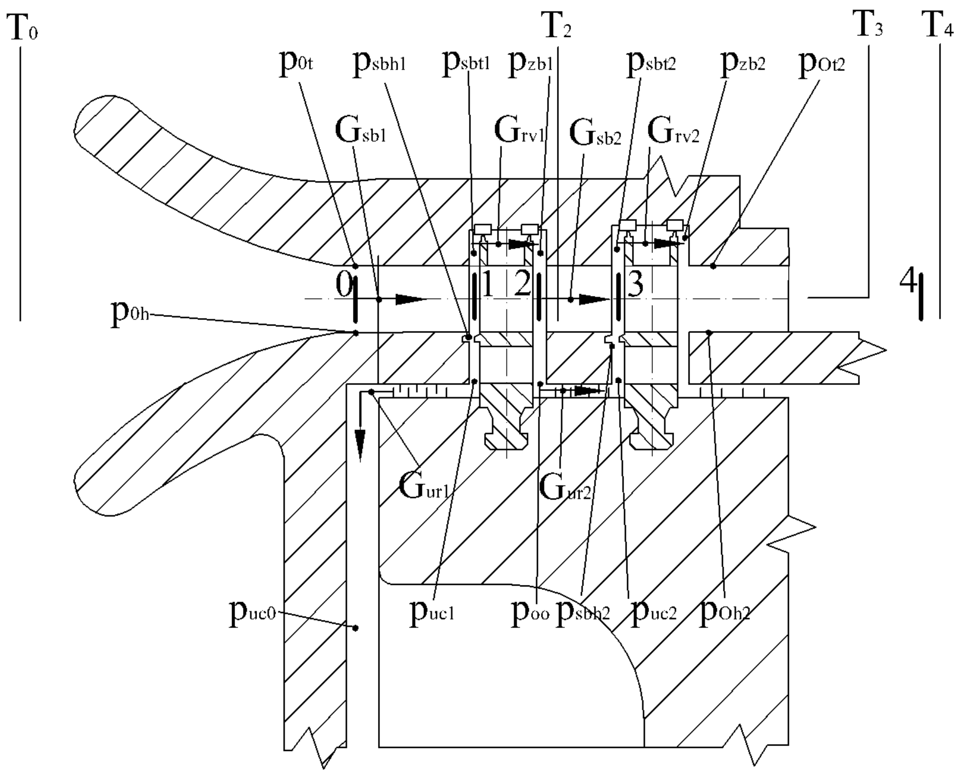

Figure 1. Pressure and temperature measuring points at the individual stages

MEASUREMENT RESULTS

The influence of the velocity ratio on the flow parameters

The number of the parameters has the influence on the thermodynamic efficiency of the turbine stages. Apart from the design, the final efficiency value is particularly affected by the level of Mach and Reynolds number as well as by the velocity ratio $(u_{1h}/c_{ST})_{avg} = \sqrt{\frac{\sum u_{ih}^2}{2h_{is}}}$ whereas u_{ih} is the circumferential velocity at the hub of the individual stages and h_{is} is the isentropic drop through both stages. The definitions of Reynolds and Mach numbers for 1st or 2nd stage, respectively, are as follows:

$$\mathrm{Re}_{ST} = \frac{c_{isST} \cdot b}{\nu} \quad [1]$$

$$Ma_{ST} = \sqrt{\frac{2}{\kappa - 1} \cdot \left[\left(\frac{p_{before\ stage}}{p_{behind\ stage}} \right)^{\frac{\kappa - 1}{\kappa}} - 1 \right]} \quad [2]$$

As the turbine operation takes place in the subsonic flow area as shown in Figure 2, it may be assumed that the influence of the Mach number on the efficiency is irrelevant. The used profiles are subsonic. The Reynolds numbers are in the area where the loss is strongly depended on them and thus they have the certain impact on the efficiency. However, their values during the experiments practically remained unchanged. The decisive influence on the efficiency course consequently has just the pressure distribution and the velocity ratios at the stages. The steam leakages through the over-shroud and the shaft seal as well as the flowing through the balancing slots are brought into use here.

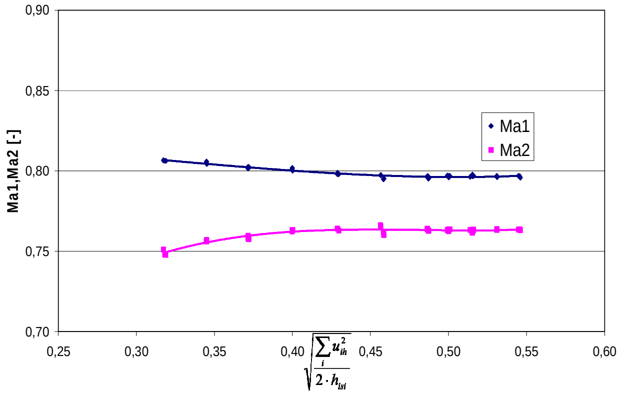

Figure 2: Mach numbers at the stages

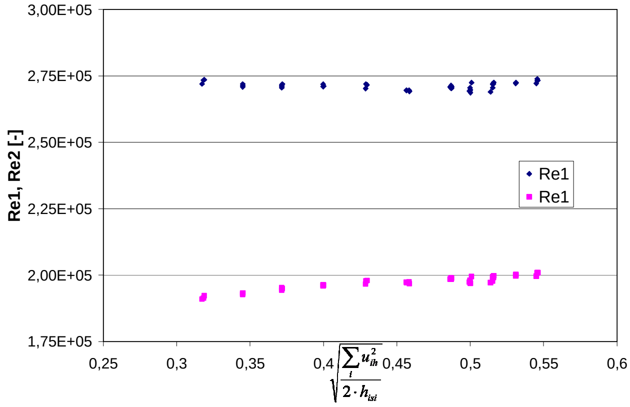

Figure 3: Reynolds numbers at the stages

The amount of the steam flowing through the turbine of the individual stages is specified by means of the condensate trapped in the measuring tank. The measurement results are stated in Figure 4. The mass flow through the stage 2 of the stator blades is decreased by the steam flow flowing through the shaft seal.

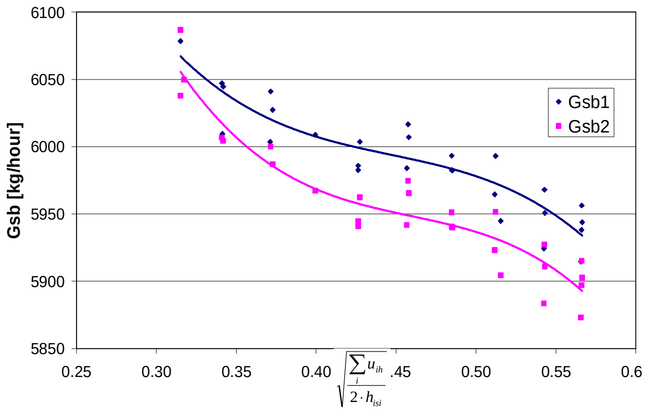

Figure 4: Steam flow through the blades

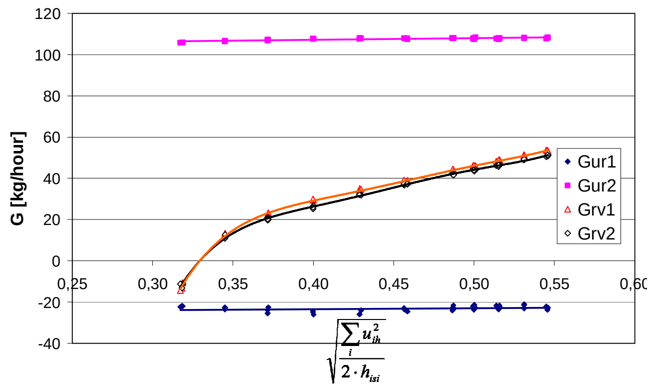

Figure 5: Steam leakage through the seals

The steam leakage through the shaft and over-shroud seals is established from the measured pressures and temperatures. Their values, depending on the velocity ratio, are stated in Figure 5. The steam flows through the shaft seals are practically constant in all operating modes. Only the steam leakage through the over-shroud seals varies. At very low velocity ratios the back flow may appear here.

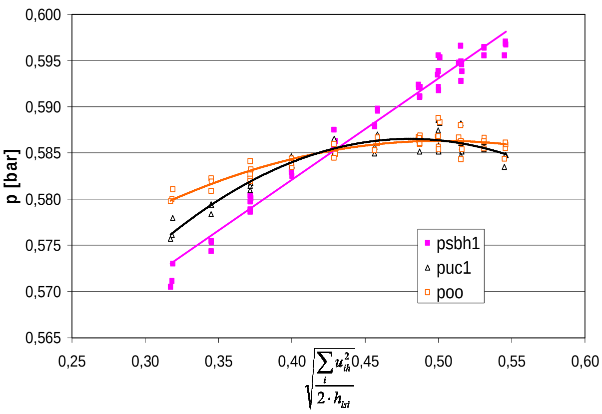

Figure 6: Pressures on the first stage slots

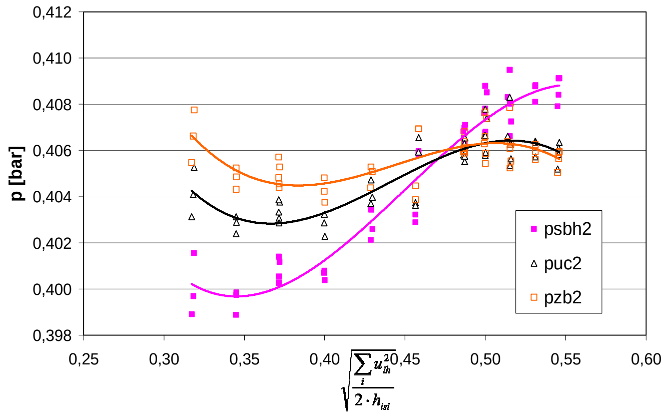

Figure 7: Pressures on the second stage slots

The steam flow through the balancing slots shall also have the influence on the efficiency of the individual stages. The explicit data have not been established for the calculation of the mass flows through the slots so far. The character of the slot flow may be evaluated from the pressures measured on both sides of the slots. The relevant pressure values are stated in Figure 6 and 7. The pressure at the hub of the rotor blades is carried out here too. The back flow on slots seems to occur under the certain conditions. The change in the characteristic pressure distribution at the first stage occurs at $(u/c)_{avg} \approx 0{,}42$ and at the second stage as late as at $(u/c)_{avg} \approx 0{,}48$. The reaction stage at the hub of the individual stages corresponds to it. The reaction represents the proportion of the enthalpy drop per a rotor blade and the total drop per a stage. Figure 8 shows the stage reaction. The blading geometry of the individual stages primarily has the influence on the reactions. Lower enthalpy drop appears to be processed at the second stage than at the first stage and consequently the actual velocity ratio is higher than at the first stage here.

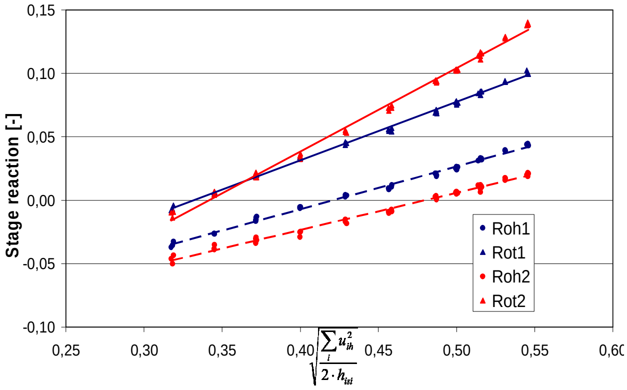

Figure 8: Reaction at both stages

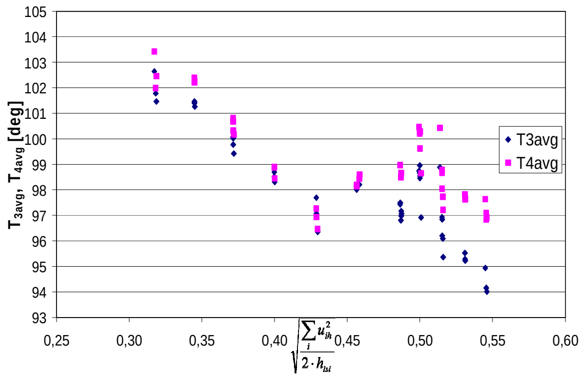

Figure 9: Steam temperatures behind the second stage

The variation in the reaction and the steam flow through the balancing slots are also evident in the course of the temperatures behind the second stage. This is presented in Figure 9. The temperature T_3 represents the local temperature in the channel centre behind the stage. The temperature T_4 is the average value from several thermometers located in an outlet chamber. It may already be affected by the steam flow through the seals and the slots. The influence of the steam used for shaft steam sealing of the experimental turbine rotor may also become evident here. Temperature T_4 is higher than temperature T_3. Both temperatures show some atypical dependencies on the velocity ratio, which is probably connected to the variation of the pressure ratios on the balancing slots. These abnormalities do not become evident behind the first stage in the course of the temperature. The variation in the flow direction through the balancing slots of the first stage becomes evident up to the second stage.

The efficiency of the individual stages may only be estimated from the course of the temperatures. However, the question is to what extent the obtained data are exact and reliable.

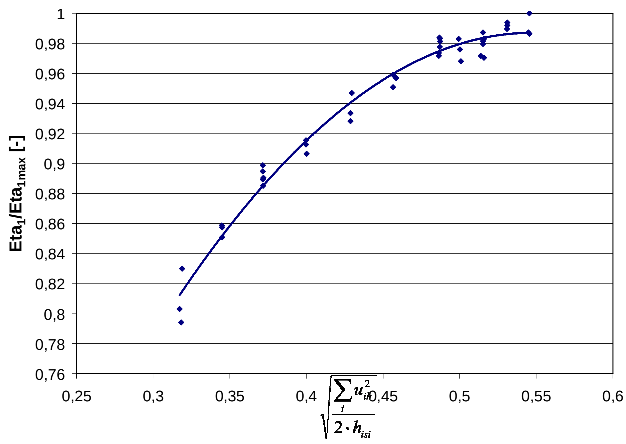

Figure 10: The temperature efficiency for the stage 1

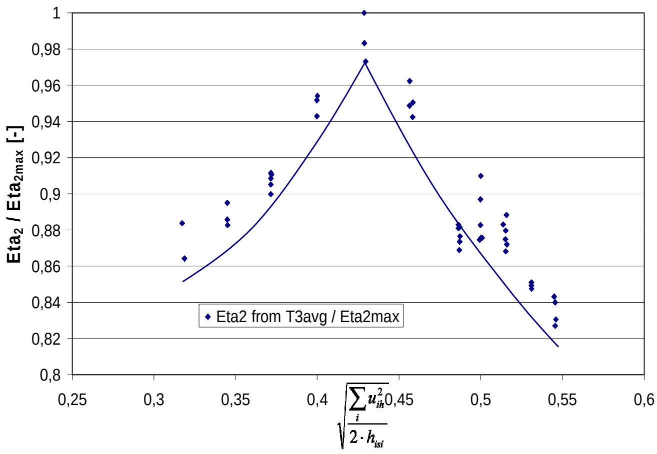

Figure 11: The temperature efficiency for the stage 2

If the efficiency of the individual stages is expressed just by means of the temperatures and/or enthalpy, obtained by IAPWS_97 wet steam functions, we get the courses shown in Figures 10 and 11. The following is valid:

$$\eta_1 = \frac{i_0 - i_2}{i_0 - i_{2is}} = \frac{f(p_0,t_0) - f(p_2,t_2)}{f(p_0,t_0) - f(p_2,s_0)},\ \eta_2 = \frac{i_2 - i_3}{i_2 - i_{3is}} = \frac{f(p_2,t_2) - f(p_4,t_3)}{f(p_2,t_2) - f(p_4,s_2)} \quad [3]$$

It is obvious that locally measured temperatures do not exactly reflect the energy conversion at individual stages and that the results here are distorted at some stages. The efficiency of the first stage may partly be improved to the detriment of the second stage. The enthalpy established from the isentropic drop (i_{2is}, i_{3is}) does not include the influence of the output stream energy that may be utilized at the contingent subsequent stages. The different enthalpy drops are processed at the individual stages. The different velocity ratios occur here too. If we carry out the partial efficiencies depending on the actual velocity ratios u/c, we get the course stated in Figure 12. It is apparent that the variation in the flow of the individual stages affects the local temperature. The efficiency established from the temperatures is not trustworthy.

The more accurate efficiency specification for both stages may just be obtained from the turbine shaft torque.

In this case the efficiency is defined as follows

$$\eta_{tv} = \frac{M_k \cdot \omega + N_l + N_V}{G_{sb1} \cdot \left(h_{is1} + \frac{c_0^2}{2} - \frac{c_{2z}^2}{2} \right) + G_{sb2} \cdot \left(h_{is2} + \frac{c_{2z}^2}{2} - \frac{c_{4z}^2}{2} \right)} \quad [4]$$

whereas M_k is the torque measured by a torquemeter, ω the angular rotor velocity and $N_l + N_V$ represents the output thwarted in the bearings and by the friction of the rotating disk surface of both stages. The other expressions arise out of Figure 1 and of the expansion process stated in Figure 13. The steam leakage through the shaft seal of the first stage G_{sb1} is negligible (G_{ur1}/G_{sb1} = -0.0033). The same mass flows practically flow through both stages. However, flowing through the shaft seal is considered at the stage 2.

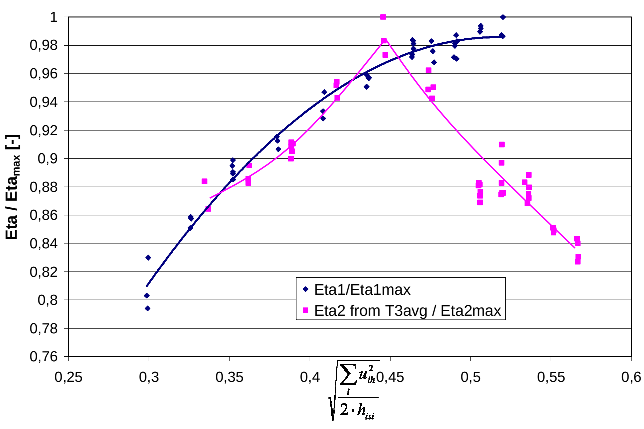

Figure 12: Efficiency from temperatures at the individual stages

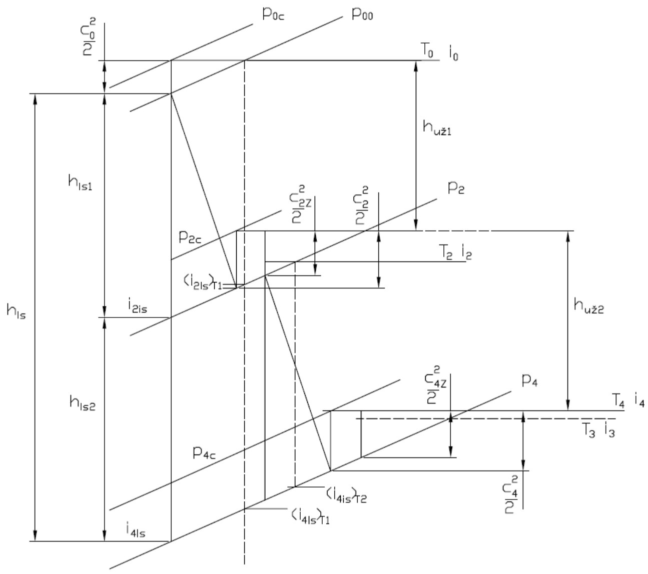

Figure 13: Expansion process at the stages

The efficiency of the two-stage turbine established from the measurement of the torque η_{tq} by a torquemeter is shown in Figure 14. The obtained shaft output and the output needed to cover the bearing and the windage losses are shown in Figure 15. The output set from the mass flow and the enthalpy drop is also considered in the diagram. The temperature T_3 should reflect the steam conditions affected by the seal steam throttling and by the disk friction in the minimal quantity. It is obvious that there are the operating conditions, whereas the losses are minimal, and the conditions correspond to the expected losses. All losses become evident by the rise of the final temperature and by the drop of the obtained output. The experiment showed that the attained efficiency level is significantly lower than the efficiency established for the same aspect ratio of the blades of the classic stage design, but operated at higher value of Re number Hoznedl et al. (2009). Consequently the efficiency decrease in this case may be primarily attributed to the influence of the lower value of Reynolds number and the steam flow through the balancing slots of the individual stages.

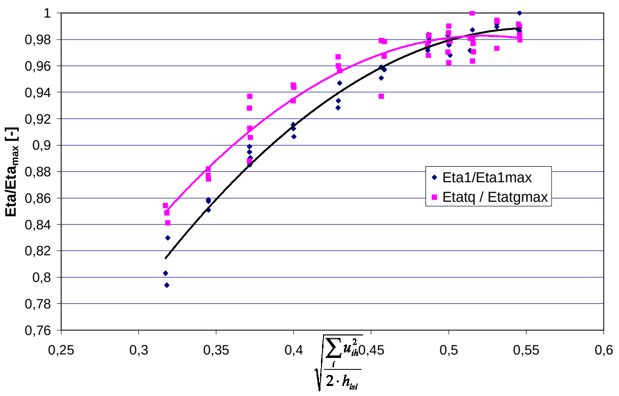

Figure 14: Comparison of the torque and the temperature efficiencies

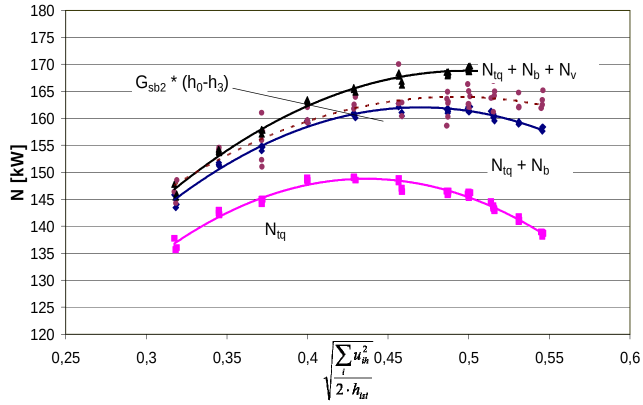

Figure 15: Shaft and blade output

Influence of Reynolds number on efficiency

In order to estimate the real impact of the low value of Reynolds number on the efficiency it was necessary to operate the turbine at its variables. The other parameters having also the influence on the efficiency such as the velocity ratio *u/c* and Mach number had to remain at the same level. The required variation was achieved by adjusting the pressure parameters at the individual stages. The pressure courses are shown in Figure 16. The average value of the velocity ratio was $(u/c)_{avg}$ = 0.505. The variation scopes of the velocity ratios and Mach numbers at the individual stages are shown in Figure 17. The variation of both parameters could not affect the efficiency of the stages. The steam leakage through the seals is proportional to the stage steam pressures as well. This is presented in Figure 18 and Figure 19 whereas the stage and the seal mass flows are stated. The attained efficiency is carried out in Figure 20. The influence of the Re number on the efficiency is confirmed. The efficiency grows with the growth of Re number too. During the experiments on the model turbine the Re number varied only within the restricted range. The areas, where the influence of the Re number on the efficiency is already thwarted, could not be reached. Based on the previous experiments (Benetka and Valenta, 2003), the efficiency growth may be achieved up to several percent.

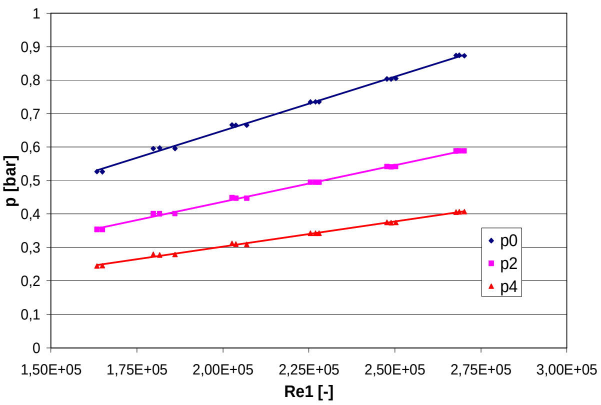

Figure 16: Pressure distribution at the stages while measuring the Re influence on the

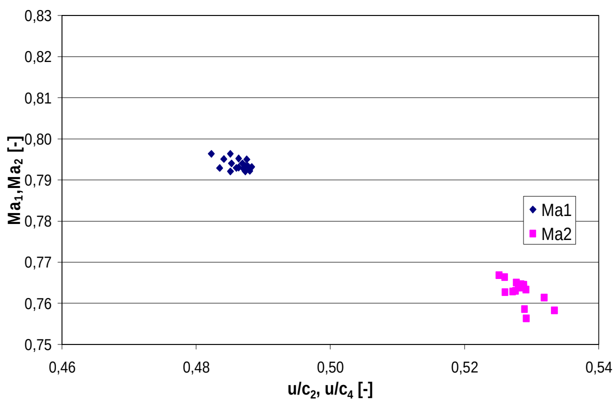

Figure 17: The variation of Mach number and velocity ratios at both stages

efficiency

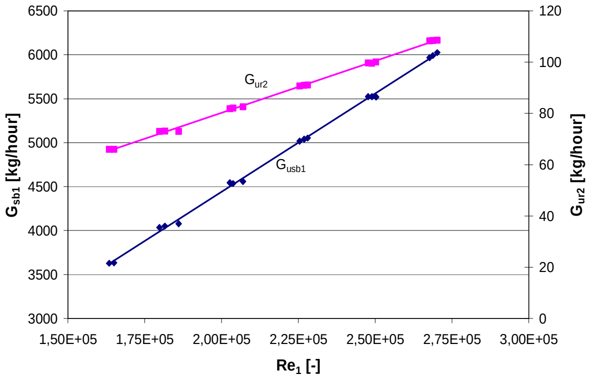

Figure 18: Stage mass flows

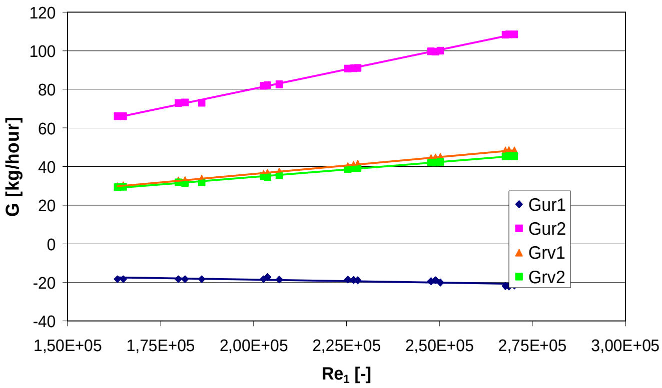

Figure 19: Steam leakage through the seals

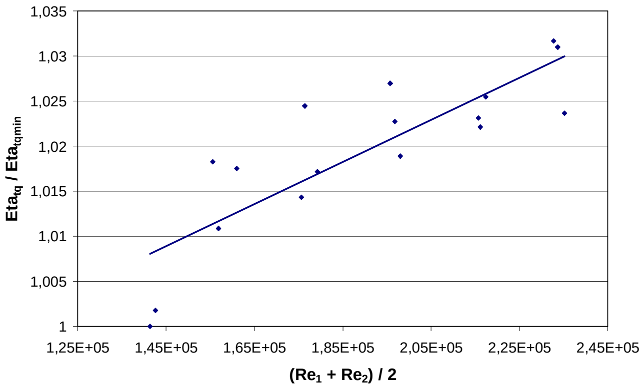

Figure 20: Influence of Reynolds number on the efficiency

Influence of the covering balancing slots on the efficiency

The resultant efficiency is certainly also affected by the steam flow through the balancing slots. It depends on the fact whether the steam at the stage 1 enters the shaft seal or is evacuated. During the experiments it was only possible to gradually cover the slots at the stage 1 and then at both stages. The variants with the steam sealing of both shaft seals and variants with the steam sealing of only the stage 2 shaft seal were always verified. The steam from the shaft seal at the stage 1 was either evacuated with G_{ur1} = -20 kg/h or maintained at the value with G_{ur1} = 100 kg/h. The natural steam flow through the seals and the slots with G_{ur2} = 100 kg/h was set at the stage 2. The final efficiency for the optimum velocity ratio $(u/c)_{opt}$ is shown in Figure 21.

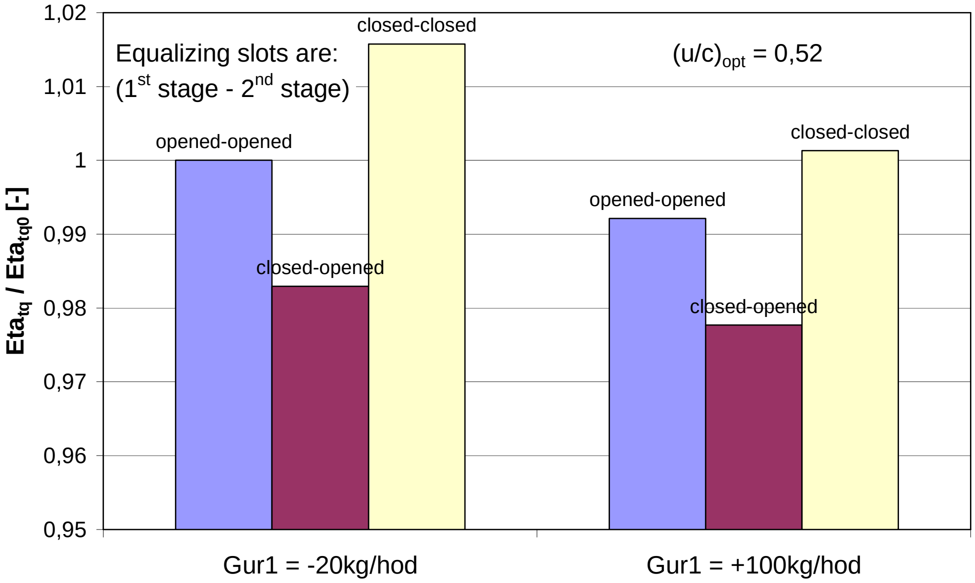

Figure 21: Effect of slot clogging on the turbine efficiency

The shaft steam sealing always leads to the efficiency decrease. The covering of the stage 1 slots causes the slight efficiency deterioration. However, covering the balancing slots of both stages paradoxically led to the efficiency improvement. The steam flow through the balancing slots may cause higher losses than the losses arisen by the steam entering the blade area from the shaft seal. It always depends on the stage response and on the fact whether the steam from the stator blade enters the slots as well or whether there is the back flow through the balancing slots. It is recommended to use the balancing slots for the impulse stages. It is preferable not to use them for the other stages.

CONCLUSIONS

- The thermodynamic efficiency of the 2-stage turbine does not correspond to the expected level for the given aspect ratio. The main cause consists in the low value of Reynolds number, which is lower by one digit position than the area of Reynolds number and by 2 digit positions lower than its operating value on the actual turbine. The flow effect through the balancing slots is also likely to be occurred.
- The efficiency for the individual stages may only be set from the temperatures measured in the centre section. The local temperature effect under the changed operating parameters should be also taken into consideration.
- The thermodynamic efficiency for the two-stage arrangement turbine is established from the rotor output, from the measured bearing losses, and from the supposed windage losses on the rotor. The proper value of the windage losses cannot be confirmed by the measurement. The available experimental data are the basis.
- The different enthalpy drops are processed at the individual stages. The different values of the velocity ratios *u/c* at the individual stages are the result. At the optimum value *u/c* of the two-stage turbine the velocity ratios of the individual stages differ from their optimum values.
- Covering only the stage 1 balancing slots causes the decrease of the thermodynamic efficiency. However, covering the balancing slots at both stages leads to the increase of the circumferential efficiency. Flowing through the balancing slots probably causes greater output of the losses than the steam entering the stage blade section from the shaft seal.
- The influence of the Reynolds number on the efficiency is substantial. The efficiency on the part may be up to 10% higher than on the tested model.

- The efficiency established from the enthalpy determined by means of the measured temperatures and pressures does not correspond to the efficiency evaluated by means of the torque on a torquemeter. The efficiency from the temperatures up to $(u/c)_{avg}$ = 0.42 increases, then it suddenly breaks and falls. At $(u/c)_{avg}$ = 0.42 the stage 1 the hub reaction changes from the negative to the positive values. Modification of the flow character through the balancing slots may occur.
- The steam temperature T_4 entering the condenser should be affected by all phenomena leading to the rise in the temperature, then by the losses on the disk as well. There is such an idea that this happens until the limit is exceeded $(u/c)_{avg}$ = 0.42, when the positive reaction at the stage 1 blading hub is attained at the same time.
- The temperature T_3 measured in the centre section right after the stage 2 should theoretically corresponds to the state involving the profile losses including the influence of the Reynolds number as well as the impact of *u/c* variation on the efficiency. Many facts show that the reality is different. The temperature T_3 is already affected by a kind of the additional loss. The findings indicate that the blade efficiency established by means of temperatures T_0, T_2 and T_3 may be untrustworthy.

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