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Two–Stage Steam Turbine under Variable Operational Parameters

Michal Hoznedl
Lukas Bednař
Ladislav Tajč
ŠKODA POWER s.r.o.
michal.hoznedl@doosan.com
lukas.bednar@doosan.com
ladislav.tajc@doosan.com

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Experimental research on steam flow through a two-stage turbine is presented. Change of pressure and velocity ratio in individual stages is studied. The effect of the Reynolds number and the Mach number on the thermodynamic efficiency of individual stages as well as the entire turbine is evaluated. Steam flows through the blades and steam leakages through the shroud sealing and the shaft sealing are described.

Keywords: Steam turbine, efficiency

1. Introduction

Thermodynamic efficiency depends on many geometrical and operational parameters of the turbine stages. There is a tendency to evaluate an effect of a single parameter while maintaining other parameters in the original condition. However, this method cannot always be applied. An experimental turbine belonging to the Doosan – Škoda Power Company has provided an opportunity to carefully examine a two-stage turbine with a drum rotor. The turbine allows variation of operational speed and pressure ratio over both stages and thus distribution of Mach numbers over the particular stages; it also allows changing the input temperature which affects the Reynolds number. As the pressures and temperatures are much lower in the experimental turbine than those in a turbine placed in a power plant, the Reynolds numbers are lower there. The turbine stages are not operated in a zone without the Reynolds number effects on the thermodynamic efficiency. Leakage of steam through the shroud sealing was monitored and its impact on the efficiency was evaluated in the studied stages. We attempted to measure how a change of the individual operational parameters affects thermodynamic efficiency over the individual stages as well as the entire two–stage turbine and to what extent steam flow through one stage affects steam flow in the subsequent stage.

2. Design of experimental turbine

The design of the turbine's flow section is illustrated in Fig. 1. The chart shows pressure and temperature measuring points and marking of local mass flows. The input steam temperature T_0 and the output temperature T_4 are measured by means of resistance thermometers located in the input and output turbine chambers, i.e. in areas with a much lower speed. The temperature on the output from particular stages T_2 and T_3 is measured by thermocouples. Basic data on blading of individual stages are given in Tab. 1. In the experiment, the clearance of shroud sealings was s = 2.23 mm.

Input parameters of the steam are kept on the specified values by means of the turbine's control system. Pressure behind the second stage is adjusted by a change of the flow area on the screen before the condenser. The rotor's output and efficiency is calculated from the torsion moment by a torque meter and/or a water break. Losses in bearings and ventilation losses are also calculated.

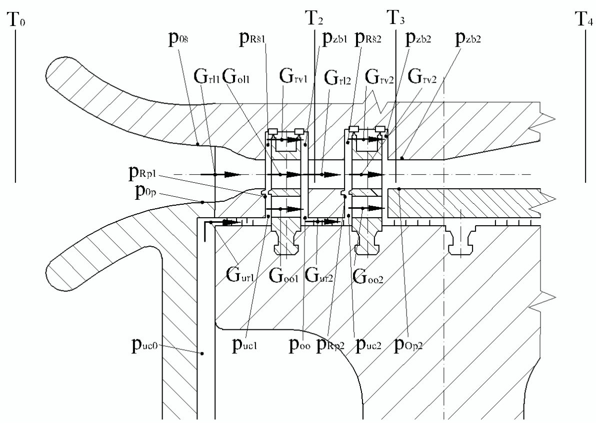

Figure 1 Pressure and temperature measuring points in individual stages

Table 1 Stages description

Description	Stage 1		Stage 2	
	Stator	Rotor	Stator	Rotor
Chord b [mm]	27	21	27	21
Length l [mm]	12	14	15.4	17.4
Number of blades z [-]	170	227	170	257
Aspect ratio l/b [-]	0.57	0.828	0.444	0.666
Proportionate pitch [t/b]	0.7	0.596	0.698	0.671

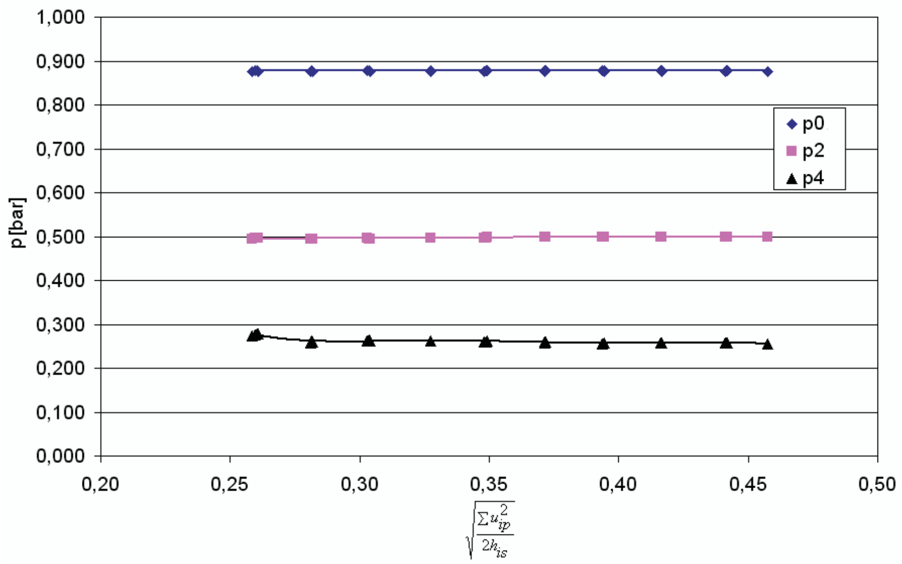

Figure 2 Pressure drops

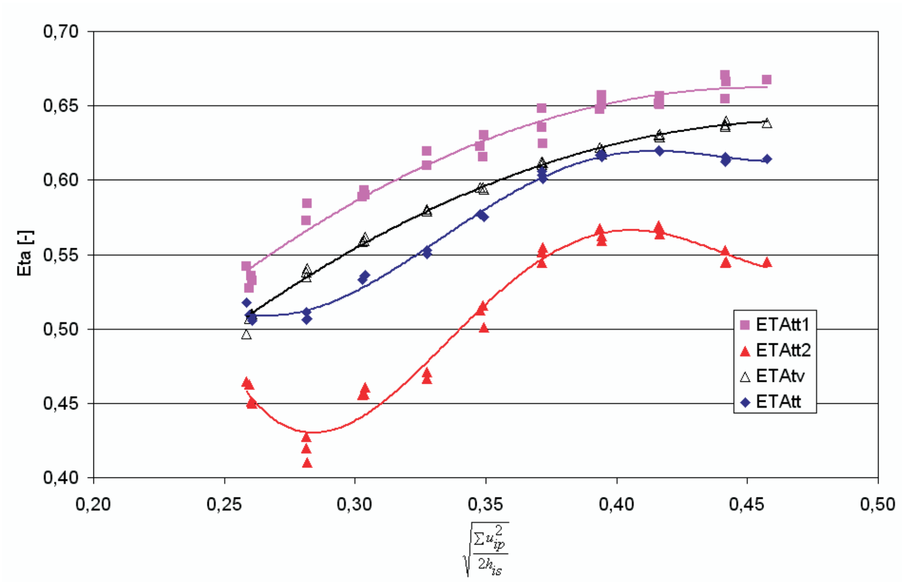

Figure 3 Stage efficiencies

3. Results from experimental research

The standard distribution of pressures in both stages is illustrated in Fig. 2. The pressure ratio is maintained constantly in both stages. The characteristic efficiency curves are illustrated in Fig. 3. The efficiency was determined from T_0, T_2 and T_4 and the respective flows. The following applies:

$$\eta_{tt1} = \frac{T_0 - T_2}{T_0 - T_{2is}} \tag{1}$$

$$\eta_{tt2} = \frac{T_2 - T_4}{T_2 - T_{4is}} \tag{2}$$

The rotor's output also includes an output spoiled on the power loss $N_l$of bearings and a ventilation loss N_v. The kinematic energy of the output steam flow $c_{4z}^2/2$ is also calculated in the resulting efficiency.

$$\eta_{tv} = \frac{M_k\omega + N_l + N_v}{G_{rl1}h_{is1} + G_{rl2}\left(h_{is2} - {c_{4z}^2}/{2}\right)} \tag{3}$$

G_{rl1} and G_{rl2} represent mass flows in particular stages and h_{is1} and h_{is2} are isentropic enthalpic flows in individual stages. Efficiency determined by temperatures is only approximate because the average value of the temperature cannot be guaranteed. In this case, the output flow energy is not considered either. The η_{tt2} efficiency is affected by a higher value of the Mach number, a lower Reynolds number and a steam flow through the shroud sealing into the main flow. Distribution of Mach and Reynolds numbers is illustrated in Fig. 4 and Fig. 5.

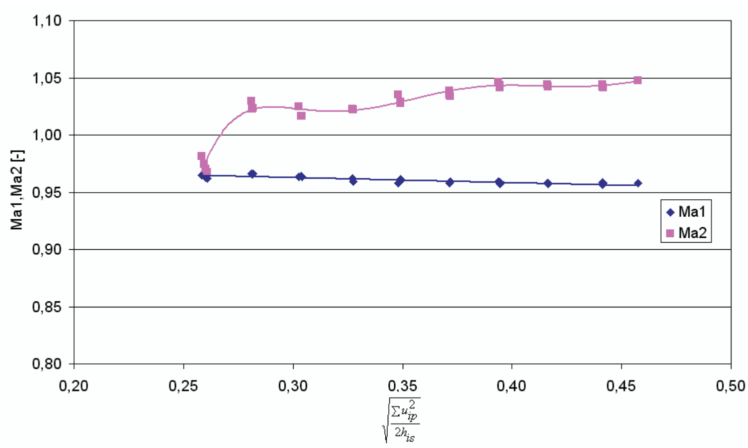

Figure 4 Mach numbers

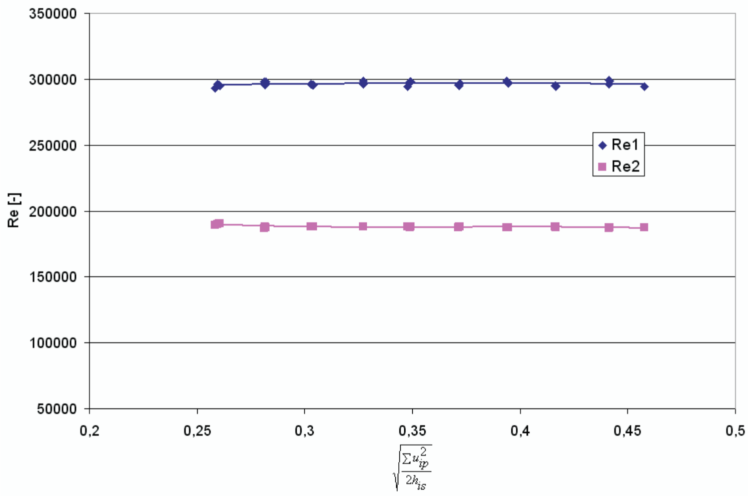

Figure 5 Reynolds numbers

It is assumed that transonic flow in the vanes of the second stage will have a radical effect on the efficiency of the second stage. To confirm this assumption, we attempted to gradually change pressure flows in both stages under a limited range of velocity ratio u/c. The input pressure can be controlled before the turbine or behind the turbine. In a two–stage turbine, the aerodynamic parameters of both stages are affected when the flow is regulated. The following charts demonstrate how the main parameters which affect the thermodynamic efficiency of individual stages gradually changed. We tested three alternatives of regulation of steam flow through the turbine. In the first alternative, the turbine was without thermal insulation and the slots of both stages were uncovered. Thus it does not represent an adiabatic process. The efficiency, the mass flows as well as the enthalpic drops in individual stages are affected. Option 2 represents a turbine with insulation when the pressure flow is controlled by changing the input pressure. In Option 3, the pressure flow was controlled by change of the pressure p_4 behind turbine. In Options 2 and 3, the slots of the second stage were covered. Fig. 6 illustrates a change of pressures on the turbine's inlet and outlet at the specified control method.

The change of pressures corresponds to the progression of steam temperatures in the individual stages. The respective data are illustrated in charts in Fig. 7. The T_4 temperature is the lowest in a turbine without insulation (Option 1). This is caused by heat transfer. In Option 2, we managed to maintain a constant input temperature. Temperatures T_2 and T_4 did not change either. The change of the Reynolds number, the Mach number and the efficiency will thus be predominantly affected by a change of pressure ratios and steam velocity in individual stages. Option 3 shows the increase of T_2 and T_4 temperature. This is a clear consequence of pressure growth behind the 1st and 2nd stage.

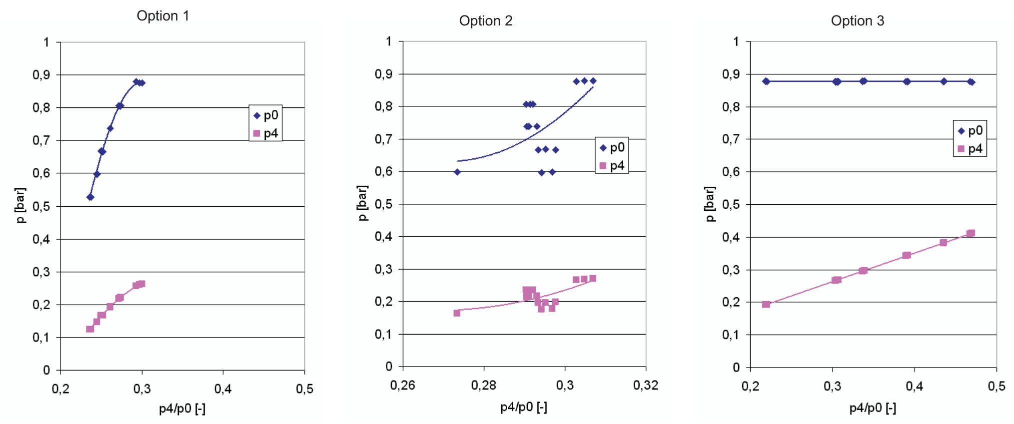

Figure 6 Pressure ratios in individual options

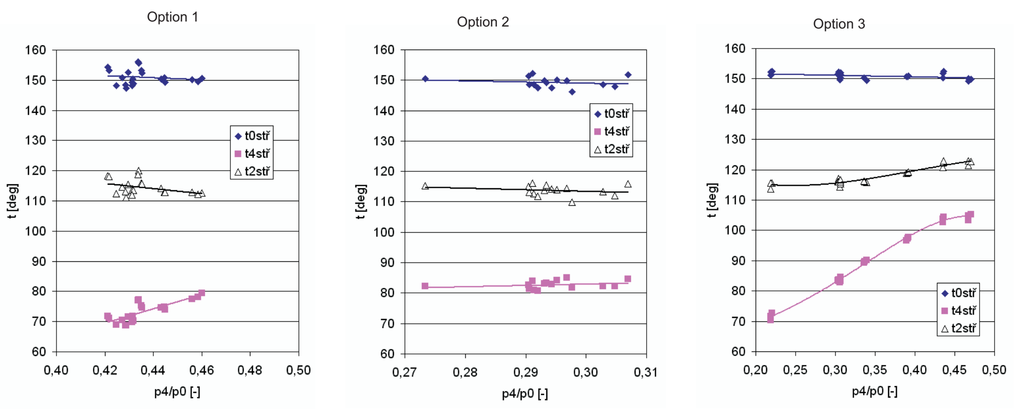

Figure 7 Curves of temperatures in the individual stages

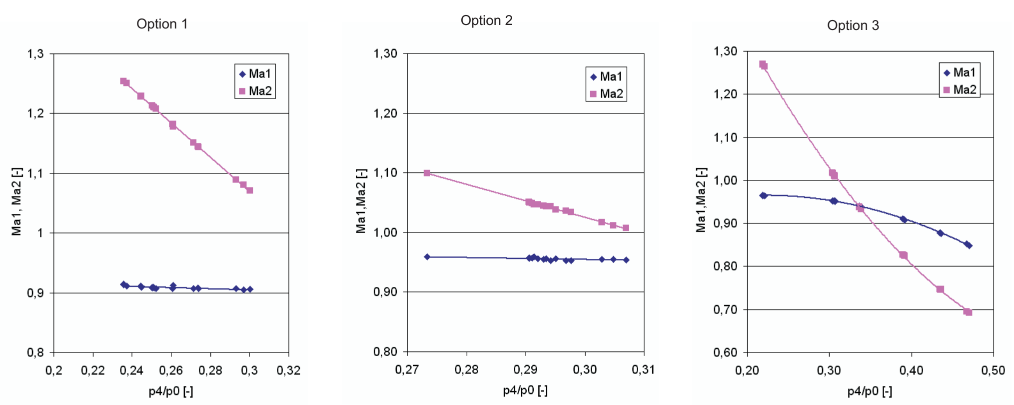

Figure 8 Changes of Mach numbers in the individual stages

Charts in Fig. 8 show how modification of pressure ratios affected the progression of Mach numbers. In the 1st stage, the Mach number is always kept on a level of transition into the transonic flow zone. Shock waves should not affect losses in the 1st stage. A significant change of Mach numbers can be observed in the 2nd stage. As it regards a supersonic flow involving Ma changes within the range of 0.7 – 1.27, additional losses should alternate within 0 – 10% on the basis of measurements in an aerodynamic tunnel. Since the experiments do not proceed in an auto model zone, the Reynolds number represents an important parameter which affects the final efficiency value. Fig. 9 gives information on the Reynolds number. The value of the Reynolds number is always lower in the 2nd stage. This is one of the explanations of lower efficiency in the 2nd stage. In Options 1 and 2 the Reynolds number grows together with the growth of the pressure ratio. As a result, the efficiency should increase in the 1st stage if the Mach number stagnates. The efficiency should also increase in the 2nd stage because the Mach number drops there simultaneously. In Option 3 the Reynolds number does not change very much; the efficiency should thus be largely affected by the Mach number and the velocity ratio.

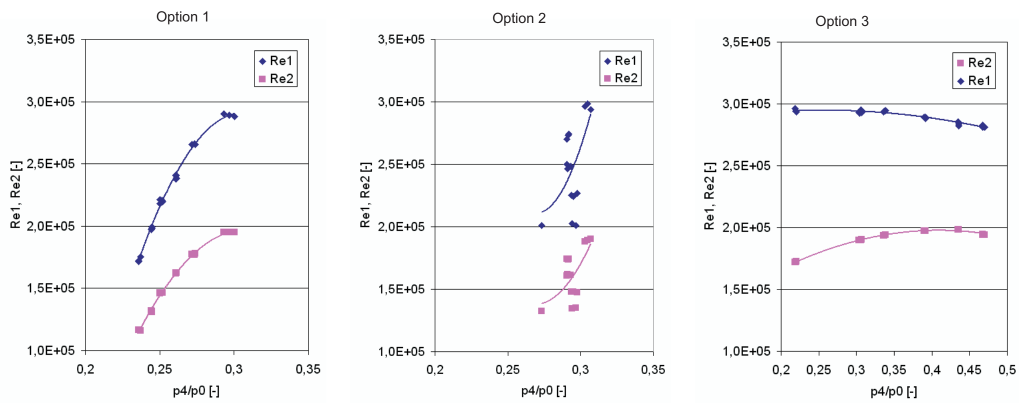

Figure 9 Changes of Reynolds numbers in the individual stages

Modifications of pressure ratios must also be manifested by the amount of steam that flows through particular stages. The mass flows through the shroud sealings will be modified as well. Fig. 10 and Fig. 11 outline the mass flows. If the input pressure grows, the mass flow of the stage also grows. In the 1st stage, the steam flow through the shaft sealing is controlled; a value close to zero is desired. Steam flow through the shaft sealing in the 2nd stage is relatively stable in all tested options. Only steam that flows through shroud sealing can have a decisive effect on losses in the individual stages. In the 2nd stage, upstream steam flow does not occur in the shroud sealing. Differences in mass flows through sealing are not very significant. Losses will only be affected by their proportional value related to the mass steam flow through the blade section of the stage.

A further parameter that can radically affect the thermodynamic efficiency of the stage is the velocity ratio u/c. Keeping this parameter within the lowest possible range is desirable. In the experiments, we monitored the average u/c value over both stages; it is specified by the Parsons number or averaged velocity ratio.

$$\left(\frac{u_{root}}{c_{is}}\right)_{avg} = \sqrt{\frac{\sum u_{ip}^2}{2h_{iS}}} = \sqrt{\frac{Pa}{2}} \tag{4}$$

However, constant u/c values for individual stages cannot be guaranteed. Fig. 12 illustrates how the u/c values changed in the various options.

If a constant Parsons number is to be maintained, then the particular u/c values for individual stages vary simultaneously when the pressure ratio in both stages changes. If the u/c value grows in the 2nd stage, the u/c value drops in the 1st stage.

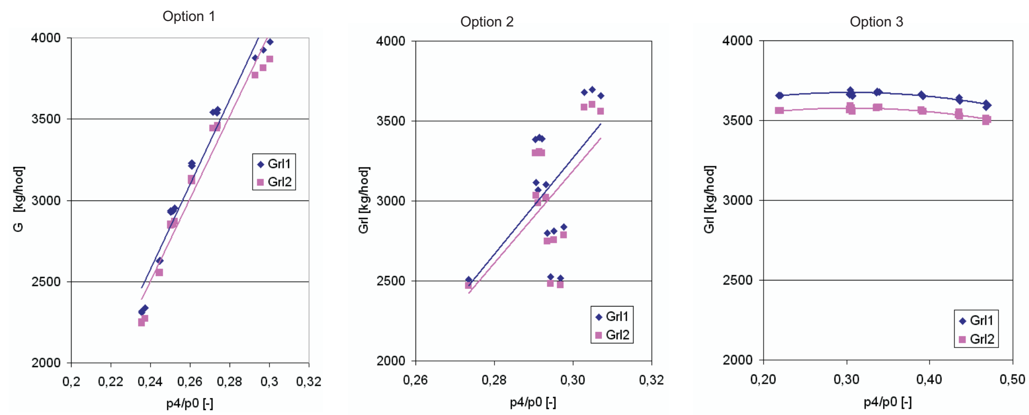

Figure 10 Mass flows through the particular stages

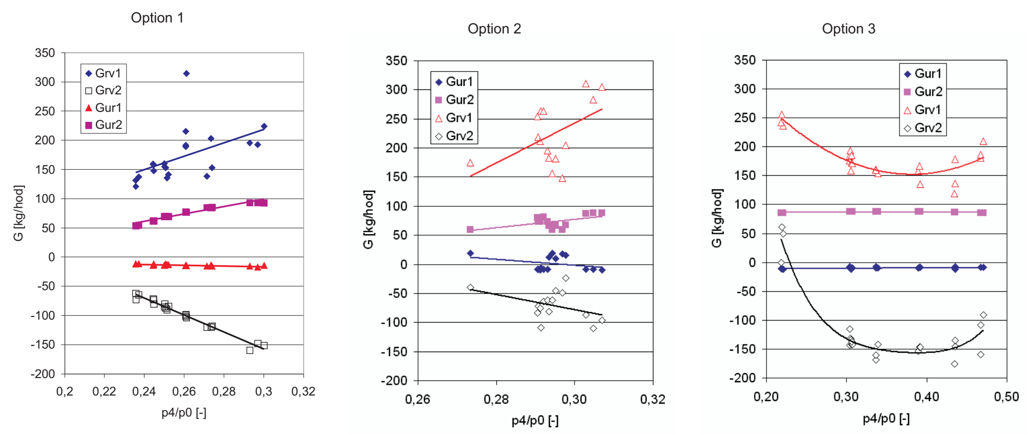

Figure 11 Steam leakage through sealings

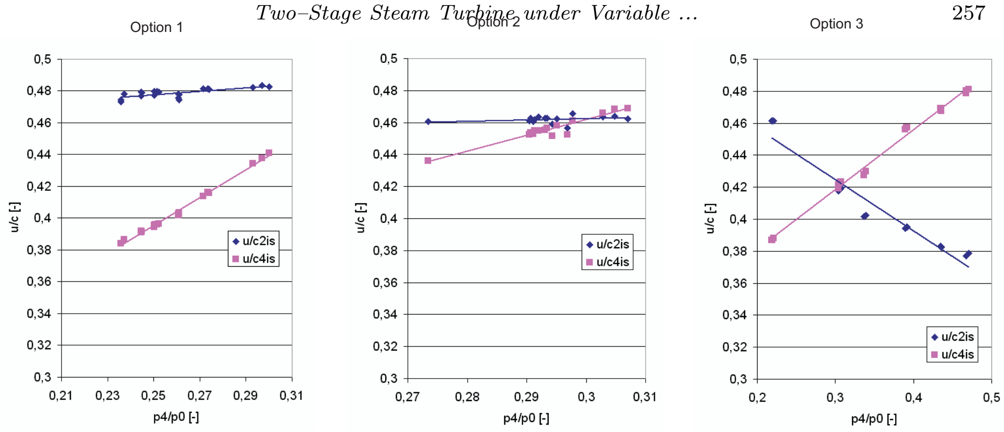

Figure 12 Curves of velocity ratios in the particular stages

This is given by the particular geometry of the flow areas, the mass flows in the stages and distribution of pressure flows over the individual stages. All rules for optimizing an impulse stage design can be applied for the 1st stage. This also regards the optimum velocity ratio value. In a subsonic flow, (up/cst)opt = 0.48. Transonic flow prevails in the 2nd stage. With regard to the T_3 temperature in the passage center behind the stage, the optimum velocity ratio equals $(u_p/c_{st})_{opt} = 0.39$. The lower $(u_p/c_{st})_{opt}$ value can also be affected by a steam flow through equalizing slots and sealings. Short blades are sensitive to a change of reaction in the radial direction. The reaction in the stage depends on the u/c velocity ratio.

Fig. 13 illustrates how the respective modifications were projected into thermodynamic efficiency determined from the temperatures. Transfer of heat from the turbine leads to an artificial improvement of the thermodynamic efficiency of the stages.

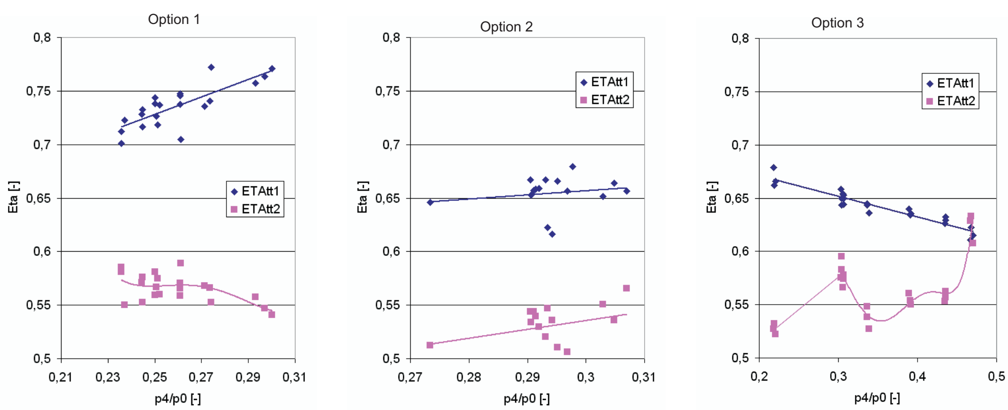

Figure 13 Efficiency determined from the temperatures

This is largely demonstrated in Option 1 of the 1st stage. The efficiency of individual stages is affected by a complex impact of all parameters described in the previous section of this article. The 1st stage should predominantly be affected by the Reynolds number and leakage of steam through the shroud sealing. The Mach number is generally steady or not very variable. The drop of efficiency in the 1st stage in Option 3 is probably caused by a drop of the velocity ratio and drop of the Mach number. The drop of Mach number, the growth of Reynolds number and the increase of the velocity ratio have a positive effect on the 2nd stage. In Option 2, the effect of steam leakage through the shroud sealing was also considered. The efficiency increases with reduction of steam loss in the 1st stage. It is concluded for the 2nd stage that the higher the upstream flow in the sealing is, the better the efficiency of the stage is.

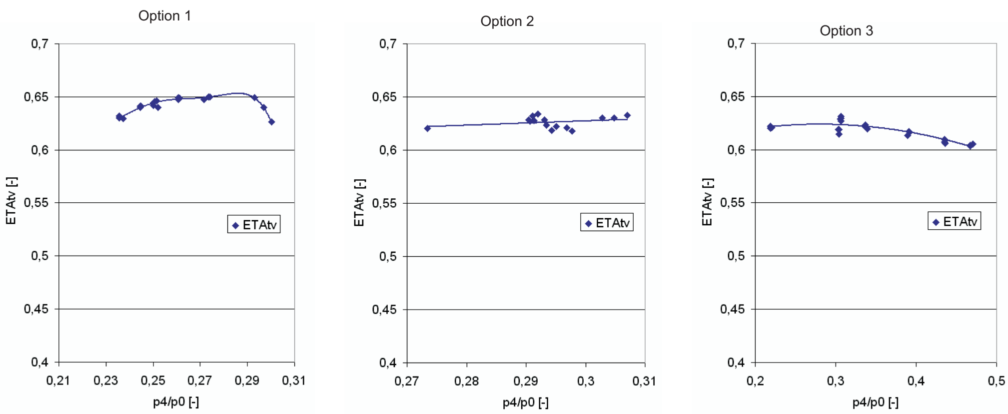

Figure 14 Thermodynamic efficiency of the turbine

Efficiency determined from the temperatures gives no information on the efficiency of energy utilization for either stage. This can be reliably determined by means of the torque moment measured by a torquemeter. Efficiency determined in this way is illustrated for individual options in Fig. 14. Roughly the same efficiency was obtained in all options for both stages. A favorable impact of the return flow in the shroud sealing of the 2nd stage and the reduced leakage of steam in the sealing of the 1st stage was confirmed. The maximum efficiency of the turbine with insulation was achieved in the pressure ratio $p_4/p_0 = 0.3$. In this case, there was a transition into the transonic flow zone in both stages. Although the efficiency $p_4/p_0 = 0.47$ determined from the temperatures for either stage in Option 3 is acceptable, the efficiency for both stages together is slightly worse.

4. Conclusions

Thermal insulation of the turbine significantly affects aerodynamic parameters of individual stages.

Efficiency determined from temperatures does not guarantee objective progression of losses in the stages. Only local temperature which may be affected by a steam flow from sealing is considered.

The efficiency of the 2nd stage is up to 10% lower than the efficiency of the 1st stage. There is an unfavorable impact of the higher Mach number and a flow through equalizing slots under the blade root.

The backflow through a shroud sealing of the 2nd stage probably has a favorable effect on the efficiency.

The individual stages are affected by a complex impact of various aerodynamic parameters. Their effect on the individual stages could not be reliably separated and operating the turbine with a single changed parameter on the experimental two–stage turbine could not be achieved.

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