17th International Conference
ENGINEERING MECHANICS 2011
Svratka, Czech Republic, 9 – 12 May 2011

EFFECT OF EQUALIZING GAPS ON THE FLOW IN THE TURBINE STAGES

K. Yun[*], Z. Jůza[**], M. Hoznedl[**], L. Bednář[**], L. Tajč[**]

Abstract: *A stage with impulse blading on a drum rotor is considered. Steam leakage through the shroud and shaft sealing has been determined by means of a numerical simulation. The effect of equalizing gaps on the stage efficiency, flow coefficient and axial force on the rotor has been proved.*

Keywords: ***Steam turbine, efficiency, drum rotor, numerical simulation.***

1. Introduction

Using a drum rotor system represents a way of improving efficiency of the turbine stages. Blades are longer and slimmer in this design. So as not to change the number of stages, the turbine runs at a higher velocity. This method requires using a gearbox between the turbine and the generator. Ventilation loss also increases. Equalizing holes for taking steam from the shaft seal cannot be used in this conception. Equalizing holes are replaced with equalizing gaps which are made on each stator blade. Numerical simulation determining the flow coefficient in the gaps and the assessment of the flow through all sections of the turbine stage was performed. The number of gaps and the influence of the number of gaps and the shape of the gaps on the axial force were assessed. The configuration of an experimental 1 MW steam turbine in the ŠKODA POWER laboratory served as a model for the calculation study.

2. CFD analysis

The calculation study considers four basic models:

Model 1 - A basic configuration of the turbine stage incorporating only the blading section of the stator and the rotor.

Model 2 - As per model 1, but with an added shroud sealing.

Model 3 - The blade section of the stage including a shroud sealing and a shaft sealing is considered.

Model 4 - Complete configuration of the stage together with seals and equalizing gaps.

Tab. 1: Stage characteristic.

Parameter	***Stator disc***	***Rotor disc***
Chord, b [mm]	27	20
Aspect ratio l/b [-]	0.57	0.87
Number of blades z [-]	170	258
Pitch to chord ratio t/b [-]	0.7	0.623

The numerical simulation was performed with the FLUENT program. In the calculation, a single-equation Spalart-Allmaras turbulent model was set up. A mesh interface together with a reference

[*] Ing. Kukchol Yun.: ČVUT Praha, Fakulta strojní; Zikova 1905/4; 16636, Praha, e-mail: kukchol.yun@skoda.cz

[**] Ing. Zdeněk Jůza, Ph.D., Ing. Michal Hoznedl, Ph.D., Ing. Lukáš Bednář and Ing. Ladislav Tajč, CSc.: ŠKODA POWER s.r.o.; Tylova 1/57, 30128, Plzeň, CZ, e-mails: ladislav.tajc@doosanskoda.com, zdenek.juza@doosanskoda.com, michal.hoznedl@doosanskoda.com, lukáš.bednar@doosanskoda.com

frame model was used for the stator-rotor transition. Boundary conditions were analogous to the experimental configuration of the turbine. They are shown in Tab. 2. A combination of two blades for the stator and three blades for the rotor was considered.

Tab. 2: Boundary conditions.

Position	***Parameters***			
	Total pressure [Pa]	***Static pressure [Pa]***	***Total temperature [K]***	***Turbulence intensity [%]***
Input	58383	-	388	3
Output	-	36904	-	3

The numerical simulation was focused on an optimum velocity ratio u/c. From the previous experiments (Ščeglajev, 1983; Škopek et al., 1988) it is known that the flow coefficient depends on the rotational speed as well as the Reynolds number. These affects were not tested in the calculation study.

3. Calculation results

The main conclusions from the calculations are described in the report (Yun et al., 2009). This article explains the impact of the factual configuration of the stage on its efficiency.

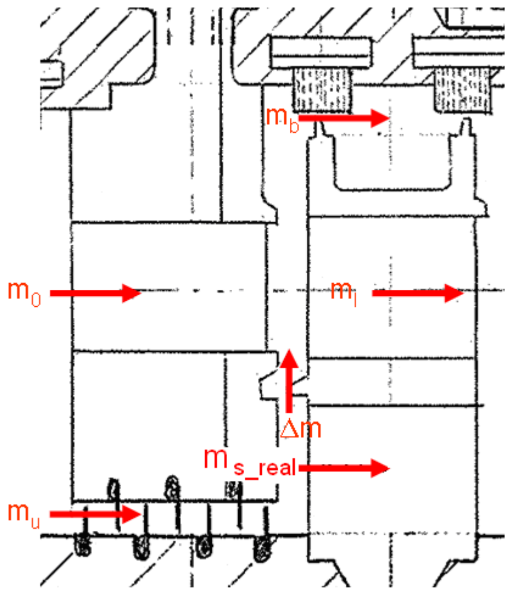

Fig. 1: Mass flows through the stages.

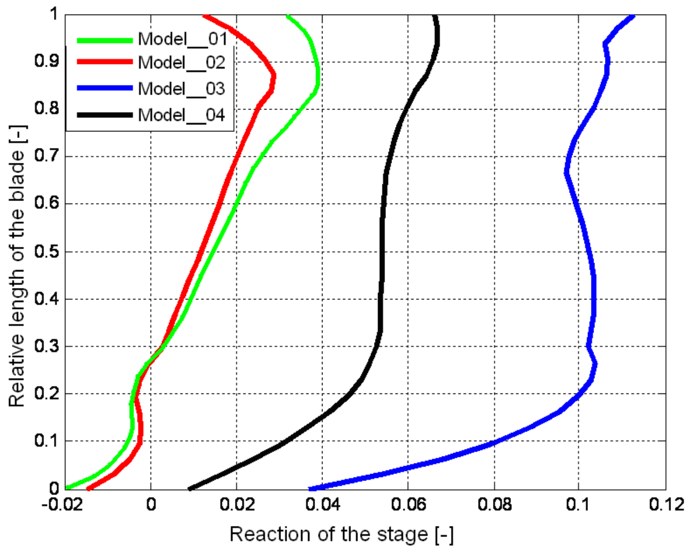

Fig. 2: Reaction curves along the blade.

The calculations allow determination of mass flows in front of individual sections of the stage. Marking of the flows is illustrated in Fig. 1. The main conclusions from the calculations for the respective models are shown in Tab. 3.

Tab. 3: Calculated flows.

Model	***m_0 [kg/s]***	***m_l [kg/s]***	***$m_u \cdot 10^2$ [kg/s]***	***$m_{s\_real} \cdot 10^2$ [kg/s]***	***$\Delta m \cdot 10^2$ [kg/s]***	***$m_b \cdot 10^2$ [kg/s]***	***μ [-]***	***η_{td} [-]***
1	0.9456	0.9456	-	-	-	-	-	0.8277
2	0.9731	0.9528	-	-	-	2.0255	-	0.7848
3	0.936	0.9568	5.6332	-	5.6332	3.5474	-	0.6732
4	0.9422	0.9381	6.4069	3.9715	2.4354	2.8493	0.0899	0.7026

The flow coefficient of equalizing hole determined from CFD is defined as follows :

$$\mu = m_{s\_real} / m_{s\_teor} \tag{1}$$

It regards the ratio of the factual and theoretical amount of the working medium that flows through the gap.

The thermodynamic efficiency has been calculated as follows:

$$\eta_{td} = \frac{a_u}{l_0} \cdot \frac{m_l - m_u}{m_c} \tag{2}$$

Here:

a_u – mechanical work,
l_0 – available energy,
m_c – total amount of operational medium,
m_u – amount of leaked operational medium without performing work.

Tab. 3 shows that even a slight slackness of the shroud sealing (s = 0.7 mm) significantly affects the efficiency. Leakage of steam through the shaft sealing should not be underestimated either. The overall decrease of efficiency is 12 %. Steam coming through the shaft sealing could not be taken away through the gaps. This is the fact that aggravates the efficiency. The flow through the shroud sealing and the shaft sealing also affects distribution of pressure heads on the rotor blade. The change of the reaction is illustrated in Fig. 2. The 01 and 02 options have a negative reaction at the root. This means that there is an increase in pressure in the rotor blade. If a shroud sealing is considered (option 2), there is a drop of pressure before the rotor blade in the tip. The shaft sealing affects the flow in the entire stage (option 3). In this case, pressure grows before the rotor blade. The equalizing gaps help reduce the overpressure on the rotor - model 04. The question is, whether it is necessary to make the gaps on each blade. For seeing of gap shape, see Fig. 3. Calculation where two gaps were considered out of three and a calculation with only one gap were made. The results are in Fig. 4 to Fig. 7.

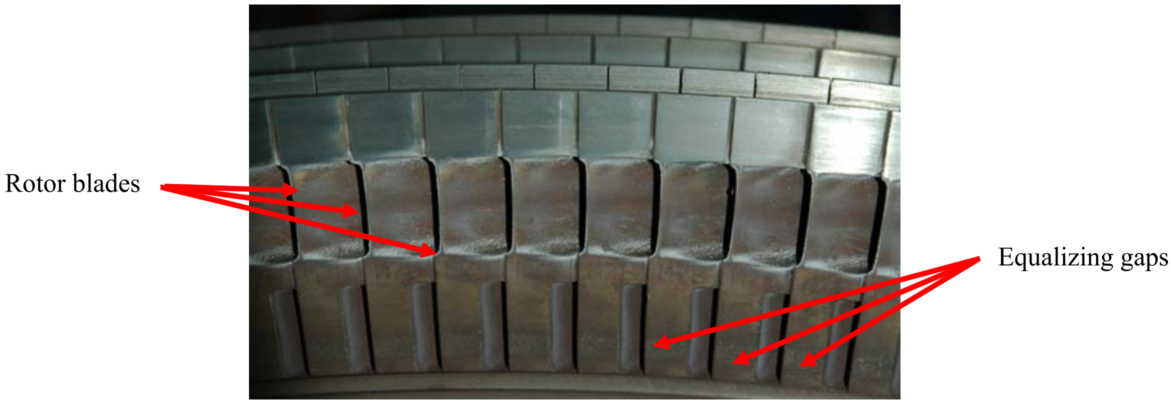

Fig. 3: Rotor blades with equalizing gaps.

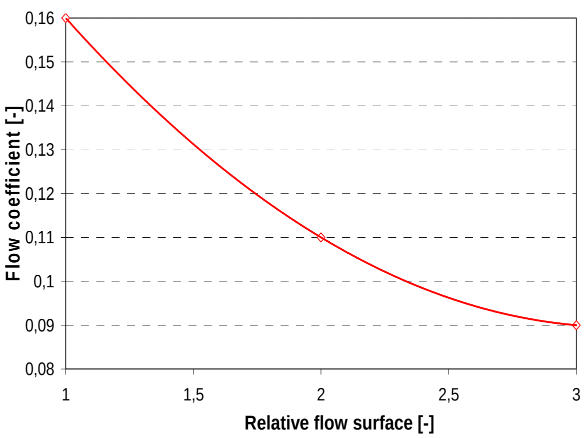

Fig. 4: The flow coefficient.

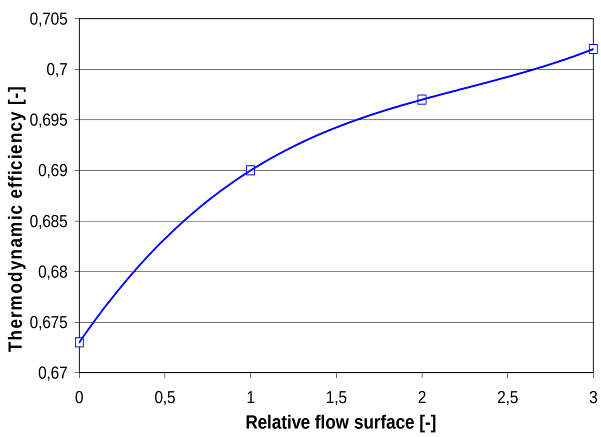

Fig. 5: Effect of gaps on the efficiency.

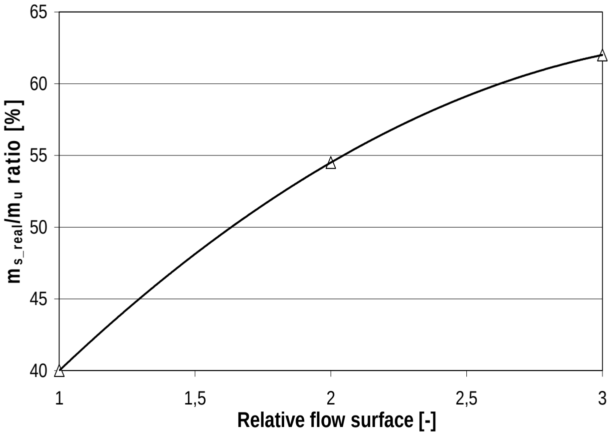

Fig. 6: Flow of the steam through the gap.

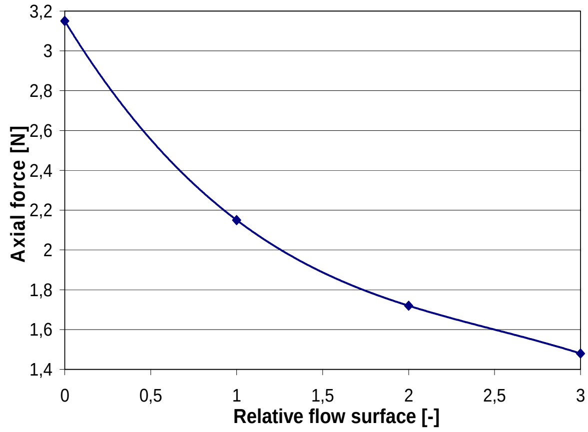

Fig. 7: Axial force on the rotor.

Value 0 on x axis in fig above means that no equalizing gap is present at this alternative. Value 1 means that one gap out of three is considered, value 2 means that two gaps out of three are considered and finally value 3 on x axis correspond to Model 4 in Tab. 3.

Clogging of the flow-surface leads to increased pressure before the rotor blade and, consequently, to an increase of the μ coefficient. The respective change is illustrated in Fig. 4. Clogging of the gaps, however, significantly affects the thermodynamic efficiency of the stage despite the growth of the flow coefficient. This is illustrated in Fig. 5. This corresponds with a lower flow of steam through the gaps, which is illustrated in Fig. 6 and the growth of the axial force on the rotor – Fig. 7. It is clear that the complete area of the gaps is insufficient. Enlargement of the area, however, is limited. The solution lies in searching for an optimum shape of the gaps so that their flow coefficient increases and in a transition to a slightly over-pressured stage. However, an increase of axial force on the rotor must be considered. There are reserves in minimizing leakage of steam through the shaft sealing, as well.

4. Conclusions

Leakage of steam through the shroud sealing and the shaft sealing affects distribution of pressure on the relatively short blades in the blade section of the stage. The shroud sealing before the rotor blade reduces the pressure; conversely, the shaft sealing increases it.

The flow coefficient has the value $\mu \cong 0.09$, which is a small value. All the steam leaking through the shaft sealing cannot just be taken away through the equalizing gaps.

The more steam gets to the rotor from the sealing, the worse the thermodynamic efficiency of the stage becomes. Clogging of the gap surfaces leads to an increase of the flow coefficient, drop of efficiency and increase of axial force on the rotor.

The solution lies in modification of the shape of the equalizing gaps, in designing over-pressurized blading and minimizing leakage of steam through the shaft sealing.

Acknowledgement

The authors of this paper wish to thank the Ministry of Industry and Commerce of the Czech Republic for the financial support of Grant No. FT-TI3/432.

References

Ščeglajev, A. V. (1983) Steam turbines, SNTL Prague.

Škopek, J., Šťastný, M., Biskup, P. (1988) Research into the Flow Coefficient in Equalizing Holes, in: Research report VŠSE, Pilsen.

Yun, K., Jůza, Z. & Tajč L. (2009) Effect of Compensating Gaps in a Drum Rotor with Impulse Blading on Flows in a Turbine Stage, in: Research report SKODA, Pilsen.