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Witold Zakrzewski[1*]
Lucjan Nastałek[1]
Janusz Badur[1]
Krzysztof Jesionek[2]
Karol Straś[2]
Mariusz Masłyk[2]

[1] The Szewalski Institute of Fluid-Flow Machinery
Polish Academy of sciences
Gdańsk

[2] Wroclaw University of Technology
Faculty of Mechanical and Power Engineering

Modeling of Baumann's turbine stage operation. Part I – Flow

A Baumann stage is one of the way, to increase the turbine's output without an increase of the last stage blade length. Due to the complicated design, the Baumann's blade technology is complex, and its efficiency is lower in comparison to a free-standing blade. Currently that stage is used mostly in back pressure cogeneration heating turbines. This paper presents the operation of low pressure part steam turbine in different conditions, calculated with two models of steam and compared to measurements and TURBINA 0D code.

1 Introduction

Huge volumetric flow rate of steam creates difficulties in designing the low pressure part of condensation turbine of large output. In the past, a technological constrains led to the two-tier Baumann stage concept with separation of steam flow onto two paths:

- internal, where steam is expanding on two stages on its way to condenser
- external, where the whole amount of available enthalpy is extracted at one stage.

*Corresponding Author. E-mail: wzakrzewski@imp.gda.pl

Each stator and rotor blade forming Baumann stage is made from one piece of metal but composes two parts partitioned with a shelf. Conic ring created from linked shelves is the flow-path separating element. Two-tier flow-path Bauman stage compared to traditional solution has few advantages:

- the two parts of the moving blade in the Baumann stage have different duties, hence there is a discontinuity in the blade profile; each part of the blade operates in different conditions and therefore flow angles can be properly adjusted,
- increased throughput capacity of low pressure (LP) part increases the power output of the entire turbine by a factor of 1.5 [1],
- the optimal condenser pressure is lower than in classical solution. Trough that the total output power is even greater,
- equipped with a Baumann stage steam turbines are smaller and lighter than traditional with the same power output,
- in the two-tier blade system, the upper blade part can be designed as low reactivity blade, therefore leakage trough blade tip clearances can be limited.

Unfortunately, the use of these basic solutions have disadvantages as well. For example, the retrofitting of an LMZ 13K215 turbine by replacing Baumann stage with one stage with longer blades, made it possible to increase the internal efficiency of the LP part by 11% [1]. Such a considerable drop in the efficiency of LP part equipped with two-tier stage is due to the following factors:

- an increased losses in the exhaust hood occurs due to the high exit velocities and complicated configuration of the flow path,
- upper part of the Baumann rotor blade operates in partly supersonic conditions,
- higher losses in last stage nozzle vane, due to an increased axial distance between the stages,
- large additional losses occur due to steam flowing from the lower tier to upper one trough the gap between stages.

There are also technological difficulties. In addition, the use of Baumann stage entails the need of using more complicated technology for making blades and more complicated methods for ensuring their vibration reliability.

Along with a lower efficiency of the LP steam path, frequent breakages of the two-tier rotating blades above partition and increased corrosion in this region, as well as at the last stage blades tips, have occurred to be the main disadvantages of this special stage. Additional investigations showed that the main origin of coarse-grained moisture causing this erosion is an intense condensation on the surface of the conical visor (detail A) separating the flow which is intensively cooled by colder expanded stem of the upper tier. Regions of increased corrosion are specified in Fig. 1.

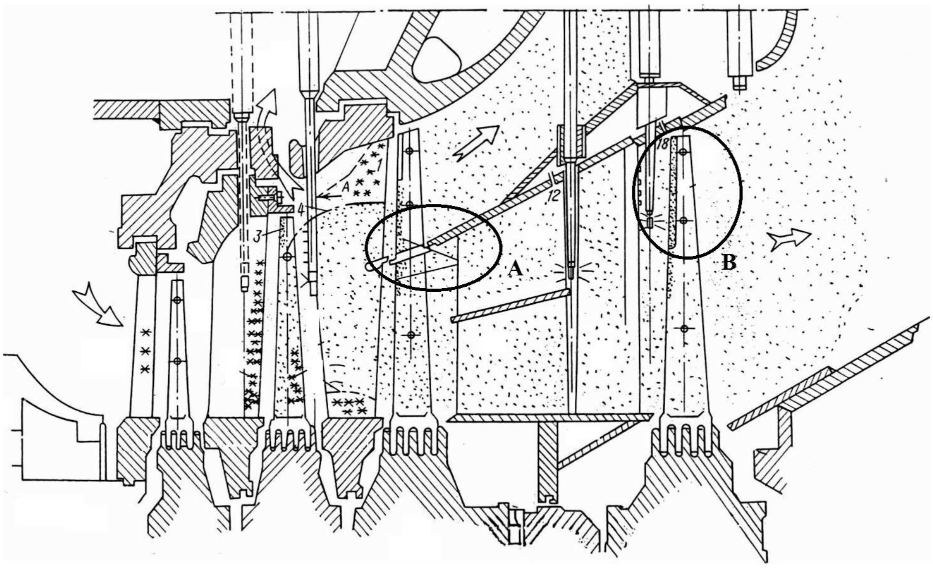

Figure 1. Low pressure part of 13K215 turbine, regions of increased corrosion: A – conical visor, B – last stage blades tips.

2 Computational model of low pressure part

The geometry of the LP part equipped with Baumann stage has been created using CAD data software. Geometric model consists of full 360 degree set of guide vanes and blades distributed in four stages, with different number of blade passages as in Tab. 1.

For better accuracy of calculations tip clearances, steam extraction after second stage, blade damping wires, as well as two-path exhaust hood have been considered in geometrical model. After discretization the number of finite volumes was beyond 13.2 millions. In the calculations whole LP part have been

Table 1. Number of blade passages.

	Guide vane	Rotor blades
Stage 1	76	124
Stage 2	64	120
Stage 3	48	92
Stage 4	48	94

considered. Some geometry details of LP part designed by LMZ (Leningradsky Metallichesky Zavod) are shown in Fig. 2.

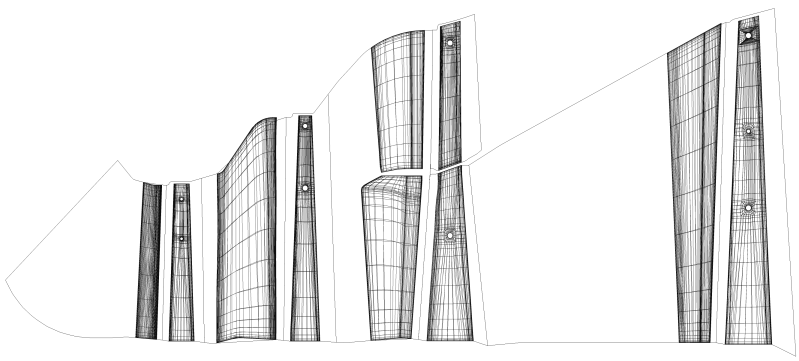

Figure 2. Numerical geometry of 13K215 LP part with Baumann stage.

To properly set the boundary conditions at inlet and outlet of the turbine, several steam data sets of LP work points were taken from measurements conducted by Energopomiar Sp. z o.o. [2,3]. This data sets are further compared to other steam models. Calculations were performed at following boundary conditions:

	Point 1 26.2 MW	Point 2 31.4 MW
• inlet pressure	$p_0 = 0.1247$ MPa	$p_0 = 0.1474$ MPa
• inlet temperature	$t_0 = 179.2$ °C	$t_0 = 181$ °C
• condenser pressure	$p_k = 0.0023$ MPa	$p_k = 0.0037$ MPa
• inlet mass flux	$m= 65.194$ kg/s	$m= 76.9$ kg/s
• extraction mass flux	$m_u = 4.621$ kg/s	$m_u = 4.2$ kg/s

In numerical calculations two models have been considered namely:

- ideal steam (without condensation)
- equilibrium condensation (with steam tables included)

Steam condensation and droplet formation occurring in LP part of steam turbine has its dual impact. Firstly droplet formation process influence overall energy conversion efficiency (energy loss on drop creation and growth) and secondly growing water droplets eventually damage turbine blades. Hence the establishing of correct method for evaluation of condensing flow through turbomachinery is extremely important. Main issue of the paper is connected with calculations of whole LP part including exhaust hood using wet steam model that gives a possibility of proper reconstruction of thermodynamic parameter across turbine.

In the equilibrium steam model which is employed in this paper, the wetness is directly (algebraically) related to the pressure and the enthalpy. As it is reported in the paper by Starzmann *et al.* [4] the mass flux difference between nonequilibrium and equilibrium case is lower than 1%, therefore it is expected that such mass flux changes lead to negligible flow velocities changes, but inlet flow angles in condensation region can change up to 5%. It is also expected that the efficiency for nonequilibrium case is lower than for equilibrium case due to the irreversible heat transfer caused by the condensation process.

For the condensation model it is assumed that, the droplets volume is negligibly small in comparison to the control volume, thus it is simply omitted. It is fully justified since droplet sizes are typically very small (0.1–100 μm). It is assumed also that there is no any interaction between droplets in the model and the velocity slip between the droplets and gaseous-phase is negligible.

To properly model the viscous fluid flow of steam, tabulated molecular viscosity was considered. Turbulent losses of momentum have been predicted with realizable k-ε model. It was found that impact of turbulent losses is relatively small. The dissipated power is on the level of 460 kW, mainly due to mass flow decrease of the value of 0.45 kg/s. All calculations have been performed with an ANSYS CFD package [8].

3 Pressure and wetness distribution

The average pressure drop for both data points, calculated by means of 0D prediction that bases on experimental data [5], ideal gas and equilibrium condensation models are presented in Fig. 3. Results indicate that the basic operational parameters as an average pressure and wetness distribution can be efficiently predicted by means of simple equilibrium condensation model. Detailed information such as subcooling level can be estimated only by employing a nonequilibrium model. In the Fig. 4 distribution of wetness level for point 1 (Sec. 2) is presented. It is shown that condensation starts after the second turbine stage. Wetness level distribution for data point 2 is quite similar to point 1.

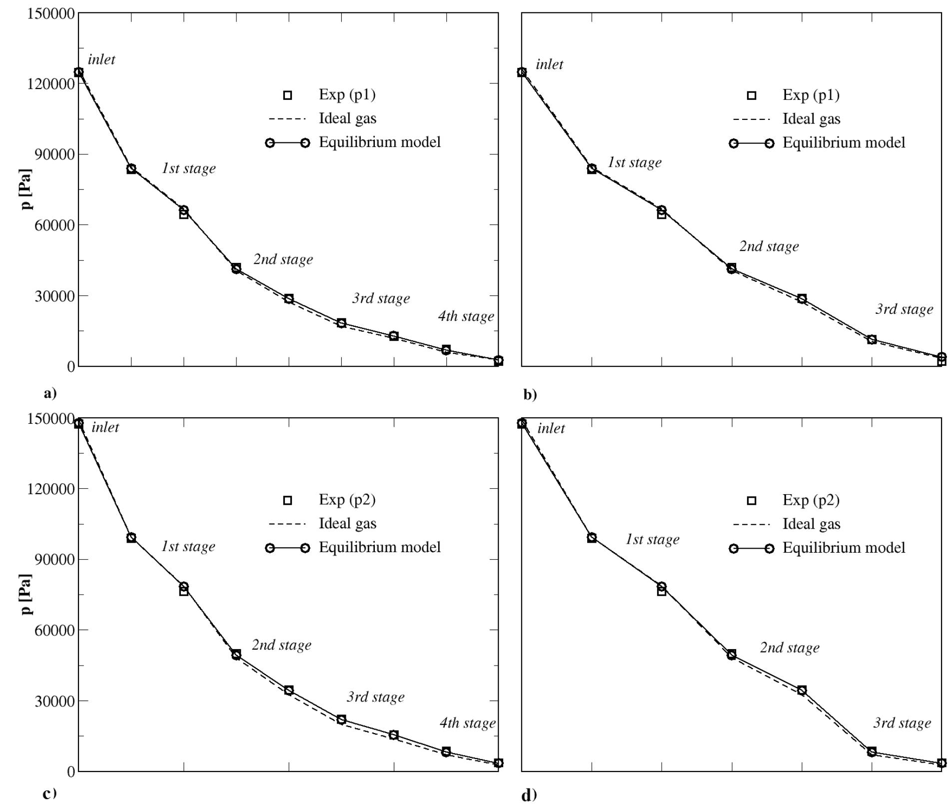

Figure 3. Pressure drop for data set point 1 throughout the: a) lower part, b) upper part, calculations for point 2, c) lower part, d) upper part.

Ideal gas model underestimates the pressure in the condensation region, while an equilibrium model provides a correct results at every turbine stage, as it can be seen in Fig. 3. Calculated results are similar to de Laval nozzle investigations [6,7], where also different steam models have been considered in order to properly predict the shift of pressure distribution.

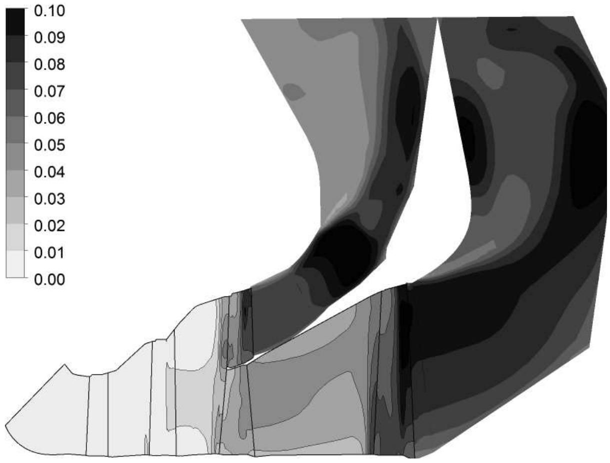

Figure 4. Wetness level (%) calculations for data point 1.

4 Power output distribution

Figure 5 indicates that the calculated power output of the first stage is overestimated at about 5% and the second at about 4% in comparison to experimental data. This can be partly explained by interstage leaks which have not been considered in our geometrical model. Leaks calculated with TURBINA 0D code [5] are respectively on the level of 5.4% and 4.6% for first and second stage.

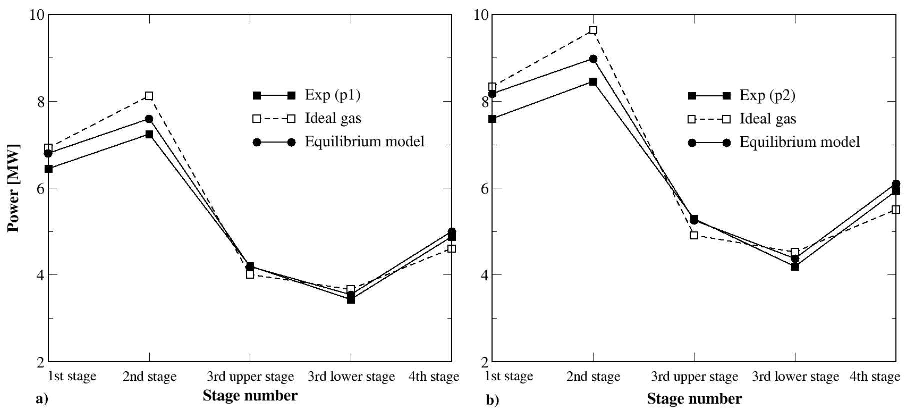

Figure 5. Low pressure part power output calculated conditions at: a) point 1, b) point 2.

5 Conclusions

Models of different kinds have been tested and compared with experimental data. Results indicate that the basic operational parameters as an average pressure distribution can be efficiently predicted by means of simple equilibrium condensation model, while ideal gas model underestimates the pressure in the condensation region. Also total power output is estimated correctly if the interstage leaks are considered. Total mass flow rate at point 1 was at the level of 65.2 kg/s and for point 2 at the value of 76.9 kg/s, this concludes that two-tier path flows can operate at very high throughput capacity at wide range of mass flow rates.

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Modelowanie pracy turbinowego stopnia Baumanna, część I – przepływ

S t r e s z c z e n i e

W pracy nawiązano do stopnia Baumanna i całej części niskoprężnej turbiny kondensacyjnej typu 13K215, w której występują różnice między pomiarami a danymi projektowymi określonymi na podstawie klasycznych narzędzi obliczeniowych typu „0D TURBINA". Dlatego celem artykułu jest próba wyjaśnienia tych różnic poprzez, dokładniejszą niż 0D, symulację 3D przepływu kondensującej pary w części niskoprężnej turbiny.