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Preliminary selection of the basic geometry of turbine stages for a supercritical turbine in a heat and power plant

The paper presents the results of preliminary design calculations of the high- and intermediate units of a supercritical turbine proposed for a heat and power plant. District water heaters connected in series are fed by the outlets of the intermediate part. A one- and two-dimensional model of flow through the blade rings of the stages were used in the design calculations. The solution resulted in the number of stages in the given turbine part, the optimum drops in enthalpy in them and the basic geometry of the stages (blade length and the distribution of geometrical blade angles along the radius). The calculation results of the heat and power plant cycle for the design conditions were the input data for the calculations of the geometry of stages. The results constitute input data to calculate the heat and power plant cycle for different operating conditions.

1 Introduction

Currently the specialized computational programs are widely successfully used (also by authors) in the simulation of energy system [1,2,3]. The use of commercial codes for the calculations of power and heat and power plants for different operating conditions does not always give satisfactory results. The reason for that is the way in which the steam parameters at turbine bleeds, which have the biggest impact on the accuracy of the cycle calculations, are determined. This especially concerns the analysis of systems with heat and power turbines, for which some stages operate in conditions much different from the design ones. For such turbines, these are first of all the stages preceding the bleed from which steam is extracted for heating purposes. This is caused by many factors which have an impact on the operating conditions of the stages, such as: the turbine-set

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electrical load, the thermal power of the heat exchanger or the desired temperature of the outlet water from the exchanger. The last value has a particularly large impact on the operation of the stages. In the cycle calculations for different operating conditions, the steam parameters at the turbine bleeds are usually determined with the use of the Stodola-Flgel dependence. It gives, as experiments have proved, fairly accurate results for groups composed of more than three stages. The Stodola-Flgel equation defines the relationship between the change in the steam mass flow through a given stage group and the change in the thermodynamic parameters before and after the group. In this method, it is necessary to know the distribution of the parameters and of the steam mass flow in the turbine flow system under reference conditions. It is usually assumed that reference conditions correspond to calculation conditions at the point of maximum efficiency. For a changed load, the pressure distribution at selected points of the turbine flow system is determined from the Stodola-Flgel equation written for individual groups of stages. In order to determine the line of expansion in the turbine, the agent temperature (enthalpy) at this point has to be determined for each pressure value obtained from the dependence. This is done on the basis of the assumed efficiency of the stage group under consideration for a given load. To evaluate the efficiency of groups of turbine stages under changed operating conditions, the dependences obtained on the basis of a statistical development of the data from a series of testing of a given turbine group are usually used in commercial codes. The testing conditions usually do not match the unique character of the operation of heat turbine stages.

As a result, any approximation to real operating conditions calls for the solution to the task of the analysis (performance of calculations) of the flow through the stages based on a 1D or 2D model for changed conditions of operation. This allows a better evaluation of the expansion line, particularly for stages preceding steam extraction for heating purposes. As the calculations presented in [4] and [5] show (Fig. 1), the stages can operate in conditions much different from the nominal ones. The solution to this problem needs, however, the knowledge of the basic geometry of the flow system of the stages. It is obtained from the solution to the design task for the stages of the turbine under analysis. The results of the solution are the number of stages in the given turbine part, the optimum drops in enthalpy in them and the basic geometry of the stages.

The aim of this study is to propose more accurate methods than those which are used in commercial codes of the thermal cycle calculations to determine steam parameters in the turbine for different operating conditions, and to couple them with the thermal cycle calculations (Fig. 2).

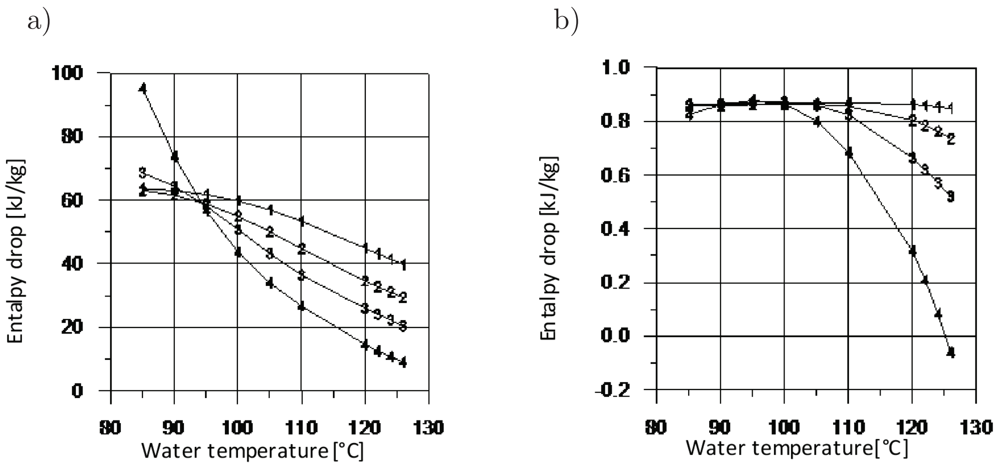


Figure 1. Isentropic drop of enthalpy in stages a) and efficiency of stages b) depending on the water temperature after the heat exchanger [4].

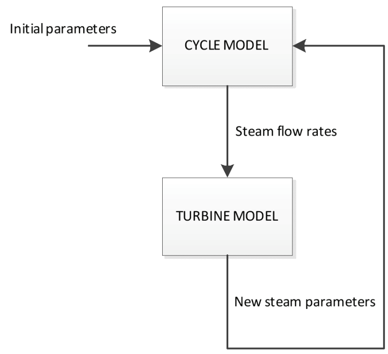


Figure 2. Scheme of the coupling of the turbine calculation model with the cycle calculations.

2 Design calculations of the stages

The authors assume that the flow parameters will be determined in control sections of the stage (the inlet, the gap between the stator and the rotor, and the outlet). Such flow is described by the following equations:

1. The radial equilibrium equation.
2. The conservation of energy equation.

3. The second law of thermodynamics.
4. The equation of continuity.

For the solving of the (design) task of synthesis, an additional condition called the "twisting principle" ("the law of twisting") is required. Most often the following twisting principles are assumed:

- the principle of constant axial component,
- the principle of constant or changeable stator outlet angle,
- the principle of constant density of the steam mass flow,
- the principle of free vortex.

The detailed equations describing the flow and the method of solving them are presented, among others, in [6–9].

2.1 Criteria for the selection of the calculation method

In tasks related to calculations of thermal cycles, one-dimensional or two-dimensional calculation methods of flow calculation through the blade rings of the stage (limited to the control sections of the stage: the inlet, the gap between the stator and the rotor, and the stage outlet) are used. The criterion for the selection of the calculation method is the diameter "d" to blade length "l" ratio:

- for $d/l > 12$ (10) – 1D calculations can be used,
- for $d/l < 12$ (10) – 2D calculations are used.

It should be emphasised that one-dimensional methods (which are much simpler than 2D methods) can be used also for stages with ratio $d/l < 12$ [10], but in that case a much smaller accuracy of the flow calculations has to be taken into account.

3 The subject of the study

The subject of the analysis is a turbine for supercritical steam parameters 30/650/670 with an electric power of 320 MW proposed for a combined heat and power plant (Fig. 3). This kind of solutions permit to achieve much higher efficiencies than in the case of currently used Polish CHP pants [10]. It is composed of a high pressure part (HP), from whose outlet steam is directed for reheat, and then to the first intermediate pressure part (IP1). After expansion in IP1, steam flows to an asymmetric double-flow intermediate pressure part (IP2 and IP3). Outlet steam from these parts feeds heating water exchangers connected in series, and two low pressure parts. Steam parameters at selected points are listed in Table 1. They were obtained on the basis of the balance calculations of the thermal cycle for operation at pure condensation.

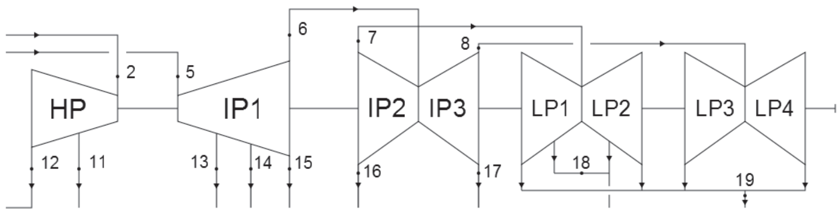


Figure 3. Diagram of the heat turbine.

Table 1. Steam parameters at selected points of the turbine.

Point number	p [MPa]	t [^{o}C]	m [kg/s]
2	30	650	225.93
11	9.666	462.9	15.94
12	6.409	402.1	210.34
6	0.584	324.7	157.3
7	0.18	196.6	72.2
8	0.09	133.6	72.2

4 Solution to the design task for the turbine HP part

In calculations of thermal cycles of power- and heat and power plants, the efficiency values for individual stage groups of the turbine have to be set. It is recommended that the experience gained from the operation of similar types of turbines and the information offered by manufactures should be used as guidelines in this matter. In the case of the bleed condensing turbine under consideration, the worst operating conditions due to flow losses will be in the high pressure part of the turbine. This is caused by high steam parameters. According to manufacturers, the internal efficiency values of the HP part can already exceed 90%. However, this concerns power units with much higher power capacities, which translates into much larger steam mass flows. The smaller the volume of the steam mass flow (which decreases as parameters rise), the lower values of the blade height should be expected. This will result in a decrease in the expansion efficiency in the stages, which applies to the high pressure part of the turbine in particular.

The potential efficiency which can be achieved by this turbine part can be estimated by solving the design task. As a result, the following are obtained:

- the number of stages,
- the diameters of the stages,
- the geometrical angles of the blades,
- the number of blades in the ring,
- flow losses in the stages.

A computational code based on a 1D model with the full modelling of flow losses was selected for the calculations [4,11–14].

The drop in enthalpy in the stages was selected due to the following condition:

$$\left(\frac{u}{c_t}\right)_{opt} = \frac{\sqrt{\eta_D}\cos\alpha_1}{2\sqrt{1-\rho}} , \tag{1}$$

where: u – tangential velocity at the mean radius, ρ – stage degree of reaction, η_D – efficiency of the stator, α_1 – stator outlet angle. Theoretical velocity c_t is related to the total isentropic enthalpy drop in the stage Δi_c by dependence $c_t = \sqrt{2\Delta i_c}$. The fulfilment of condition (1) results in the stage maximum peripheral efficiency defined as:

$$\eta_u = \frac{l_u}{\Delta i_c - \mu\dfrac{c_2^2}{2}} = \frac{c_1\cos\alpha_1 - c_2\cos\alpha_2}{\Delta i_c - \mu\dfrac{c_2^2}{2}} , \tag{2}$$

where: l_u– stage peripheral operation, c_1 – steam stator outlet velocity, c_2 – steam stage outlet velocity, α_2 – velocity vector angle c_2, μ – outlet energy utilisation factor in the next stage.

The lengthening of the blades can be achieved by reducing the rotor diameter. This, however, involves a rise in the number of stages, which may have an adverse effect due to the increase in the rotor length. An adoption of a bigger diameter results in a shortening of the blades (which is not advantageous in terms of efficiency but results in a fall in the number of stages, which may have a beneficial effect on the structure). This is presented in Fig. 4 where top line represents a mean diameter for solution with 29 stages and bottom line represents a mean diameter for solution with 33 stages. The results of the design calculations for the variant with a smaller number of stages are listed in Tabs. 2 and 3 and presented in Figs. 4–7.

Table 2. Thermodynamic parameters – pressure, temperature and velocities of steam at control sections of selected stages.

Stage number	p_0 [MPa]	p_1 [MPa]	p_2 [MPa]	t_0 [°C]	t_1 [°C]	t_2 [°C]	c_1 [m/s]	w_1 [m/s]	w_2 [m/s]	c_2 [m/s]
1	29.4	28.8	28.2	648.4	644.4	640.9	118.6	33.0	121.1	31.8
2	28.2	27.7	27.1	640.9	637.1	633.5	119.2	33.2	121.0	32.6
3	27.1	26.6	26.0	633.5	629.8	626.2	119.8	33.3	121.5	32.8
14	16.6	16.2	15.8	547.8	543.4	539.4	127.9	35.9	127.0	34.7
15	15.8	15.5	15.1	539.4	535.1	531.1	128.0	35.8	127.1	34.7
16	15.1	14.7	14.3	531.1	526.8	522.6	128.4	35.9	129.7	35.8
27	7.9	7.7	7.4	430.4	425.3	420.2	140.8	40.4	141.7	40.8
28	7.4	7.1	6.9	420.2	414.9	409.9	142.8	41.2	142.0	40.7
29	6.9	6.6	6.4	409.9	404.5	399.5	143.5	41.3	142.7	40.9

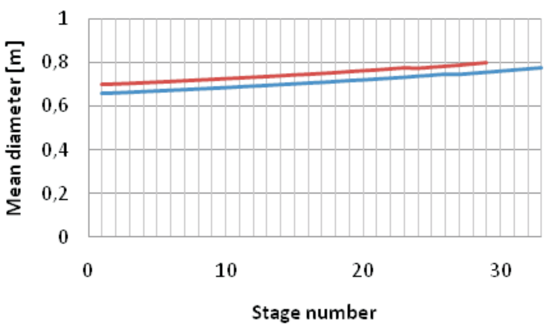


Figure 4. Distribution of the rotor diameters in the analysed variants of the design calculations.

The enthalpy drops in the HP part blade rings, which were selected due to the highest efficiency, change relatively slightly (Fig. 5). The internal efficiency

Table 3. Steam mass flows and main geometrical dimensions of selected blade rings of the stages.

Stage number	m [kg/s]	d [m]	l_1 [m]	l_2 [m]
1	221.729	0.698	0.041	0.044
2	221.729	0.70	0.042	0.044
3	221.729	0.703	0.043	0.045
14	221.729	0.74	0.057	0.06
15	221.729	0.744	0.059	0.062
16	221.729	0.747	0.061	0.063
27	206.34	0.791	0.083	0.085
28	206.34	0.797	0.086	0.089
29	206.34	0.803	0.09	0.093

of the stages which corresponds to the drops is defined by the following formula:

$$\eta_i = \eta_u - \zeta_p - \zeta_t \ , \tag{3}$$

where ζ_p – loss related to steam leakage through the ring sealing, ζ_t – loss related to the friction of rotating components.

The internal efficiency value is the lowest for the first high pressure part stage and it rises in subsequent stages (Fig. 6). The lower efficiency of the stages on the inlet side results from bigger boundary losses, and especially from higher leakage losses for these stages compared to the final ones. Boundary losses are related to the blade length – the shorter the blades, the higher the losses. The leakage losses are an important item in the balance of the losses in the turbine stage. The mass flow of the leakage depends on the sealing structure and, first of all, on the steam parameters. In turbines for supercritical steam parameters, the density of the agent at the inlet to the HP part is the highest, which is related to big leakages in these stages. The power capacity of the stages is determined on the basis of the defined enthalpy drops in the stages and their efficiency (Fig. 7).

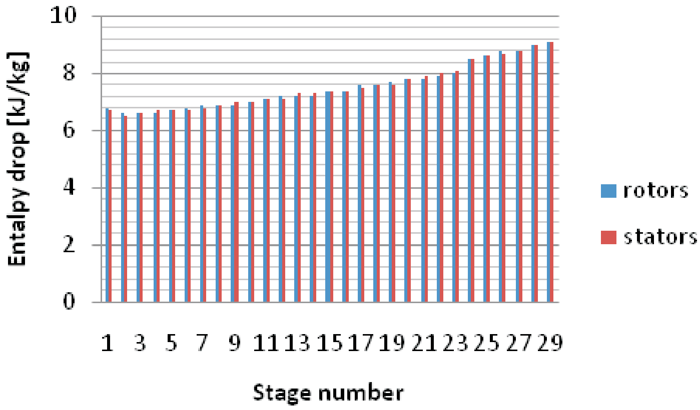


Figure 5. Distribution of enthalpy drops in the rings for the 29-stage variant.

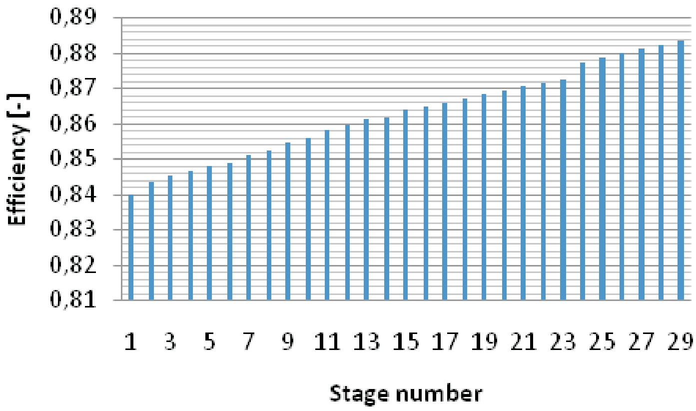


Figure 6. Internal efficiency of stages for the 29-stage variant.

5 Internal efficiency of stage groups

On the basis of the flow calculations it is possible to determine the internal efficiency of stage groups in the turbine HP part (two groups, Fig. 1). These efficiency values constitute the input data for the calculations of the cycle.

The internal efficiency of stage groups is defined by the following dependence:

$$\eta_i = \frac{i_1 - i_2}{i_1 - i_{2s}} \tag{4}$$

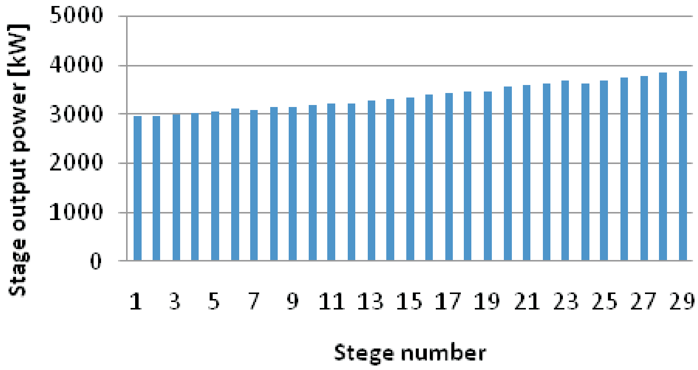


Figure 7. Internal power capacity of stages for the 29-stage variant.

where i_1 – enthalpy at the inlet to the group, i_2, i_{2s} – enthalpy at the outlet from the group for the isentropic and real expansion, respectively. For the first stage group, the efficiency should be defined precisely. The isentropic enthalpy drop can be determined both as H_{i1} and H_{i2} (Fig. 8). The difference is the consequence of accounting for the pressure drop measured before the turbine (usually before the shut-off valves) and at the inlet of the first ring. Even if the turbine operates with sliding pressure control (the control valve fully opened), the drop has to be taken into account in accurate calculations. This is confirmed by the results of the efficiency calculations for the HP part stage groups for the design heat turbine with electric power of 320 MW. They are presented in Table 4. For the assumed two-percent drop in pressure of the steam in the inlet pipe (from the shut-off valve to the inlet to the 1 stage), the difference in efficiency is 1.37 percentage points. This is an element that should be taken into account in the calculations of the cycle.

Table 4. Efficiency of stage groups of the turbine HP part.

Internal efficiency:	Value
the flow system of the 1 stage group	0.8716
the 1 stage group; pressure losses at the turbine inlet taken into account	0.8579
the 2 stage group	0.8777

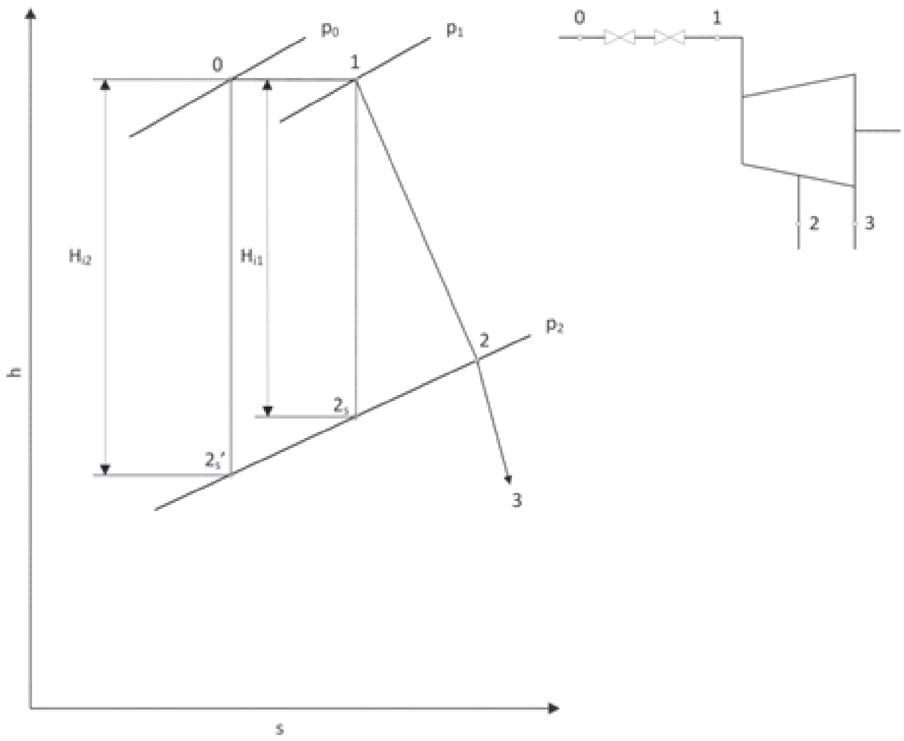


Figure 8. Expansion line in the turbine; pressure drop in the steam supplying valves – taken into account.

It should be noted that the efficiencies of stage groups are higher than those of the stages that the groups are composed of. This is due to a partial recovery of friction heat.

Another factor which has an impact on the accuracy of the calculations of the cycle is taking account of external leakages in the turbine. In the case of the reaction turbine structure under analysis, the HP part rotor has to have a balance piston. Part of the steam mass flow getting into the turbine will flow through the piston sealing, causing a reduction in the mass flow expanding in the turbine flow system $\dot{m}_{up} = \dot{m} - \dot{m}_p$ (Fig. 9). Due to the high parameters of the steam at the inlet to the turbine and the steam small specific volume related to them, the mass flow of the leakage may be relatively large. If the use of a labyrinth seal with 50 blades and a 0.0005 m gap is assumed, the leakage through the piston will be of approx. 7 kg/s, which is approx. 3% of the mass flow feeding the turbine. The mass flow in the flow system will be reduced by the leakage value, which will not only affect the turbine power capacity, but also change the parameters in the regeneration system.

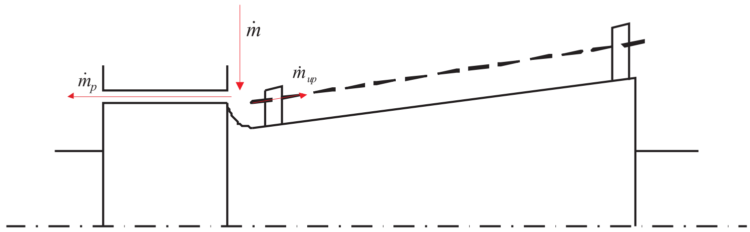


Figure 9. Diagram of steam leakages in the rotor of the reaction type structure.

6 Calculations of the intermediate pressure part

The preliminary geometry of this turbine part was determined for the turbine operation for pure condensation. The parameters at the inlet and at the outlet, and the steam mass flows in parts IP2 and IP3 were obtained from the design calculations of the cycle for the conditions of pure condensation (Tab. 1). The calculations were conducted for different shapes of a meridional flow passage (Fig. 10). A two-dimensional flow model was used in the calculations. The outlet parameters from one stage constituted the boundary conditions for the calculations of the next stage.

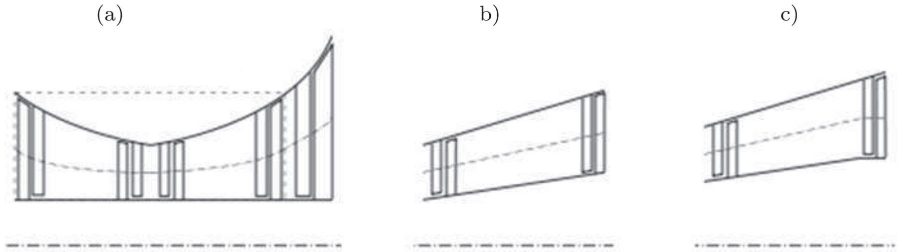


Figure 10. Diagrams of the IP part flow systems.

Selected results of the calculations are listed in Tab. 5 and presented in Figs. 11 and 12. The calculations were conducted with the assumption that the stage degree of reaction at the mean radius was 0.5 and index u/c_t is determined by Eq. (1). Radial distribution of the reaction and velocity in the first and last stage (part IP3) is presented in Fig. 11.

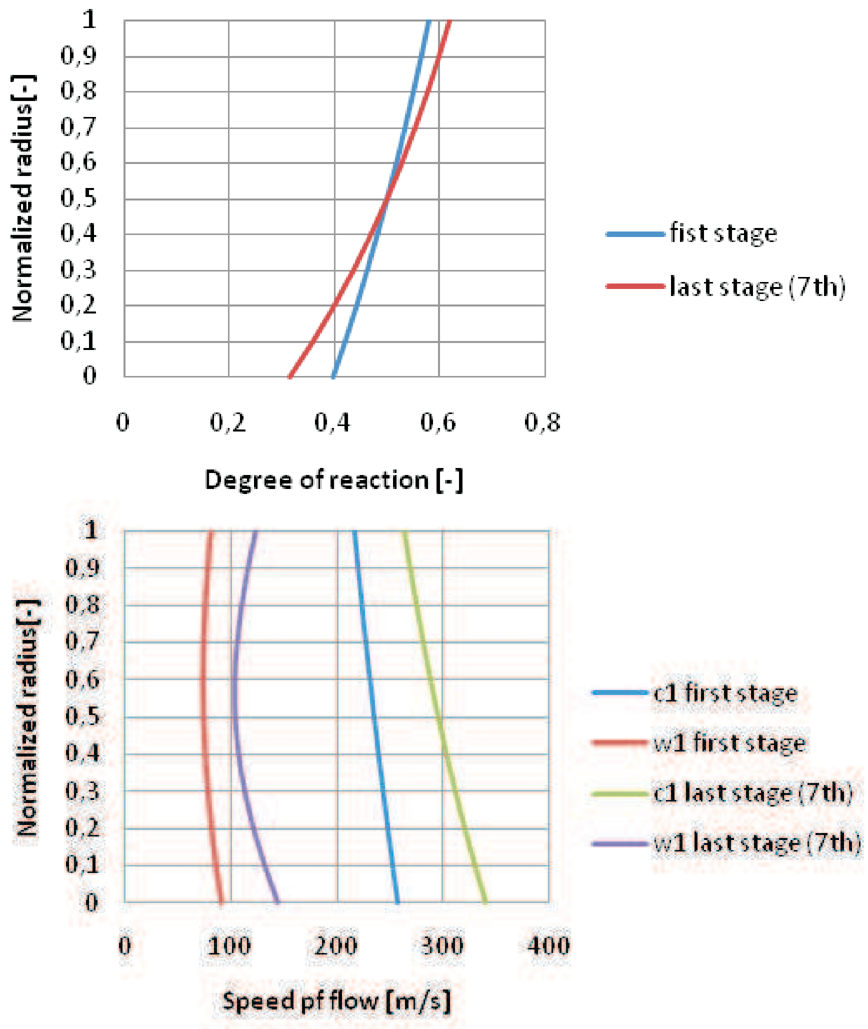
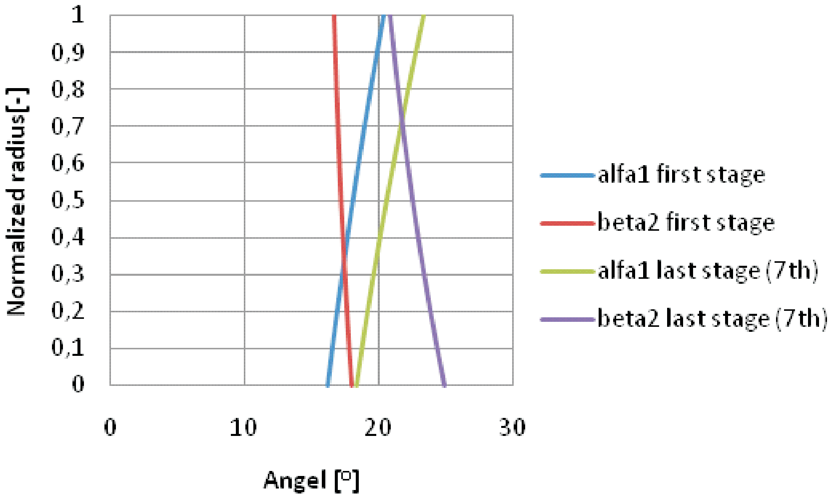


Figure 11. Distribution of the degree of reaction and velocity in stages 1 and 7.



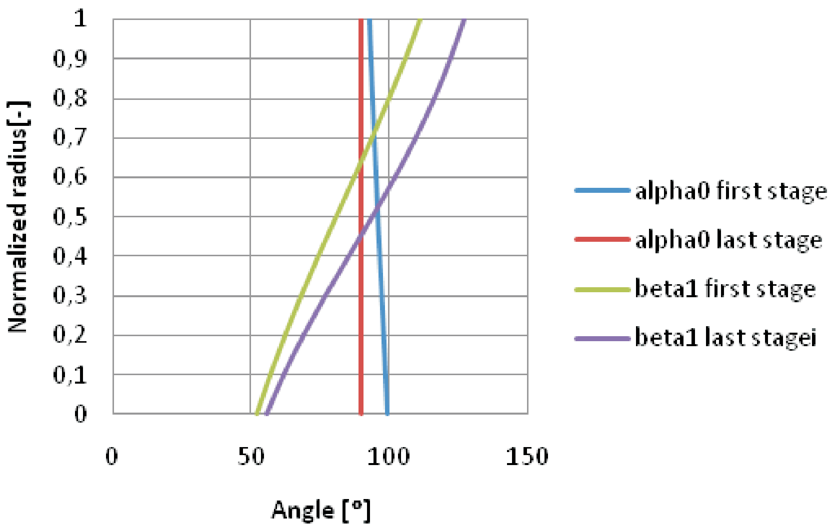


Figure 12. Distribution of geometrical angles of blades for stages 1 and 7.

The determined pressure distribution in the turbine intermediate part is presented in Tab. 5. For the flow passage marked as variant **a**, the pressure after the fifth stage is slightly higher than the IP2 outlet pressure, while after the eighth stage it is a little lower than the IP3 outlet pressure given in Tab. 1. The change of the flow passage (variant **b** and variant **c**) resulted in a rise in enthalpy drops in stages. The pressure after the fifth stage is in this case lower than in Variant a, but after the seventh stage already, the pressure is lower than the IP3 outlet pressure adopted in the calculations of the cycle.

The basic geometry of the blade rings of the turbine stages obtained from the design calculations makes it possible to solve the task of the flow analysis for this turbine for different operating conditions.

Table 5. Distribution of pressure after the stages of the IP2 and IP3 parts of the turbine.

Stage number	Pressure after the stage in [MPa]	
	Variant a in Fig. 10	Variant b and variant c in Fig. 10
1	0.4858	0.4785
2	0.3999	0.3847
3	0.3243	0.3033
4	0.2587	0.2344
5	0.2018	0.1733
6	0.1554	0.1196
7	0.1179	0.0766
8	0.0878	

7 Conclusion

While designing the cycles of a heat and power plant with turbines for ultra-supercritical steam parameters, it is necessary to perform an analysis of the potentially obtainable efficiency of individual stage groups of the turbine. It is connected with the much lower electric power than that of the turbines for the condensing power units currently designed and built. This applies to the high pressure part of the turbine in particular. The stages of this part, due to the small volume of the steam mass flow, have short blades, which causes higher losses in these stages.

According to the results of the conducted calculations of the flow in this turbine part, the internal efficiency of the 1 stage group will slightly exceed 87%, and with the pressure losses at the inlet taken into account - it will be by approx. 1 percentage point lower. The efficiency of the 2 stage group will reach approx. 88%. Considering the progress both in the aerodynamics of blade cascades and in the sealing structure, the efficiency values may be higher.

The results of the calculations of the intermediate part show that the design calculations of the cycle should be coupled with the design calculations of the turbine. This will allow an appropriate selection of the pressure values at the turbine bleeds.

The calculations of the flow in the turbine will also make it possible to take account of a series of factors, such as leakages, which, as the calculations have shown, will have an essential impact on the results of the cycle modelling. The factors should be taken into consideration when the decision concerning the final selection of the cycle structure is made.

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References

[1] Bartela Ł., Skorek-Osikowska A., *Ekologiczny efekt sprzężenia nadkrytycznego bloku węglowego z instalacja turbiny gazowej.* Rynek Energii **2**(87), 2010, 8–13 (in Polish).

[2] Skorek-Osikowska A., Bartela Ł., Model kotła oxy na parametry nadkrytyczne - analiza wybranych parametrów. Rynek Energii, 5(90), 2010, 69–75 (in Polish).

[3] Badyda K., Kupecki J., Milewski J., *Modelowanie hybrydowych układów energetycznych bazujących na procesie gazyfikacji węgla.* Rynek Energii **3**(88), 201074–79 (in Polish).

[4] Łukowicz H.: *Zadania analizy w obliczeniach przepływowych turbin parowych w zastosowaniu dla diagnostyki i projektowania.* Zeszyty naukowe Politechniki Śląskiej Number 1699, Gliwice, 2005 (in Polish).

[5] Łukowicz H., Łukowicz P., Bartela Ł.: *Steam turbine model for simulation of work under changing condition.* In: Proceedings of ECOS 2011, 705–718.

[6] Sirotkin J. A.: Aerodinamiczeskij *rasczet łopatok osiewych turbomaszin.* Mashinostroienie, Moscow 1972 (in Russian).

[7] Chmielniak T., Łukowicz H.: Rozwiązanie odwrotnego zagadnienia dla przepływu pary mokrej w stopniu. Zeszyty Naukowe Politechniki Śląskiej, ser.: Energetyka **60**, Gliwice 1977, 17–33 (in Polish).

[8] Chmielniak T., Łukowicz H., Misiewicz A.: Zagadnienia optymalizacji stopni turbinowych z łopatkami zwijanymi. In : Proc. 4th International Science and Technology Conference: "High power capacity steam turbines", Elbląg-Gdańsk, 1988, 358–367.

[9] Perycz S.: Turbiny parowe. Ossolineum, 1992 (in Polish).

[10] Chmielniak T. Łukowicz H.: *One-dimensional formulation of a parametric optimization problem for turbine stages.* In: Problems of Fluid-Flow Machines. Wydawnictwo IMP PAN, Gdańsk 1993,. 147–161.

[11] Zaporowski B.: *Analiza efektywności energetycznej i ekonomicznej skojarzonego wytwarzania energii elektrycznej i ciepła w elektrociepłowniach parowych oraz gazowo-parowych.* Ryenk Energii **5**(78), 2008, 41–44 (in Polish)..

[12] Szczeglajew A. W.: *Steam Turbines.* Energia, Moscow 1976 (in Russian).

[13] Aleksejeva R. N, Bojcova E.A.: *Pribliżennaja mietodika opredielenija aerodynamiczeskich potier w wiernych resztkach turbiny stupiueni.* Teploenergetika **12**(1973), 21–25 (in Rusian).

[14] Traupel W.: Thermische Turbomaschinen, 1 Band, 1997.

Wstępny dobór podstawowej geometrii kanałów przepływowych stopni turbiny pracującej w układzie elektrociepłowni na parametry nadkrytyczne

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

W pracy przedstawiono wyniki wstępnych obliczeń projektowych wysoko i średo-prężnych części turbiny proponowanej dla elektrociepłowni. Połączone szeregowo wymienniki wody ciepłowniczej zasilane są parą z upustów części nisko-prężnej turbiny. W obliczeniach wykorzystano zarówno modele jedno jak i dwu-wymiarowe. W wyniku obliczeń uzyskano liczbę stopni w poszczególnych częściach jak i optymalne spadki entalpii oraz podstawowa geometrię stopni (długości łopatek oraz wielkości kątów łopatkowych wzdłuż promienia). Warunkami brzegowymi do zrealizowanych obliczeń były wyniki obliczeń bilansowych bloku elektrociepłowni. Natomiast wyniki obliczeń projektowych turbiny posłużyły do obliczeń bloku przy zmiennym obciążeniu.