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Polish aspects in the modern design solutions of steam turbine flow systems

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Abstract

This paper presents new solutions of turbine stages which have been applied to high-power steam turbine flow systems. The structures, elaborated in Poland, were designed with the use of the authors' original computational software and verified by measurements performed on real turbine sets.

Keywords*: steam turbine, power cycles, flow systems, efficiency, errosion.*

Introduction

Most steam turbines operating in Poland were produced by Zamech Works, Elblag. For power industry needs domestic solutions were elaborated and also licence designs, mainly from Russia, were adapted. For past years it was managed to elaborate a relatively modern production process based on own computer design systems. The programs have been based first of all on co-operation with Polish scientific research institutes. It would appear that after taking over the Zamech Turbines Division by the ABB, ALSTOM, the worldwide recognized concerns, realization of Polish designs will be eliminated. In line with the new philosophy the works in Elblag took over immediately the most modern design and technological solutions of the concern and started to use foreign numerical calculation programs. However the fact of keeping Polish engineers still at work in the enterprise, has resulted in that Polish turbine solutions have been still coming into being. In this paper are described two design solutions elaborated in co-operation with the Fluid Flow Machinery Institute, Polish Academy of Sciences, Gdansk. They deal with new turbine stages: high-loaded ones and those before regenerative extraction.

Examples of realization of new stages of steam turbine

High-loaded stages

Impulse stages intended for operating under high loads, which ALSTOM, Elblag, has applied to 7CK-65 turbine set in the Zielona Gora Electric Power Station, can be deemed the best example of the modern solutions presently realized in Poland. Their superiority over other solutions of the concern has consisted in that the blading system of the turbine of 65 MW electric power and 135 MW heat power, could be accommodated in one common cylinder, that is schematically shown in Fig. 1 [1].

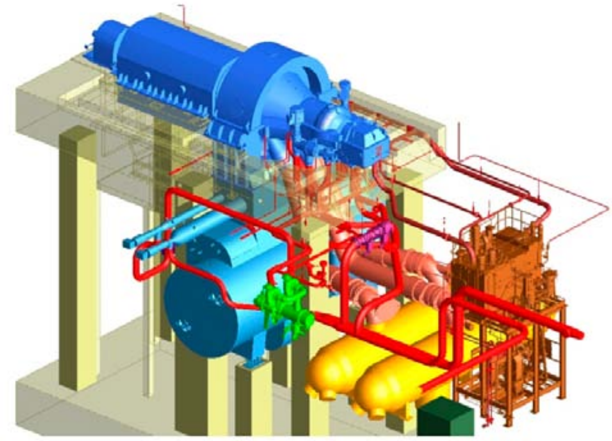

Fig.1 *Machinery devices of 7CK-65 steam turbine unit installed in Zielona Gora Electric Power Station [1].*

It was achieved at maintaining its high efficiency, improved availability, and obviously, material savings. The minimized number of 23 stages resulted from the application of new profiles permitting to operate under high loads, to majority of rings of guide vane and rotor blades. Geometry of the profiles had to be changed the higher loads to be realized. They had to guarantee a little changed kinematics of steam flow as compared with conventional solutions. This results from a change of inlet and outlet angles as shown in Fig. 2, acc. Yamamoto [2].

As can be observed in Fig. 2 the rotor blade profiles of highly loaded stages make flow passages longer,

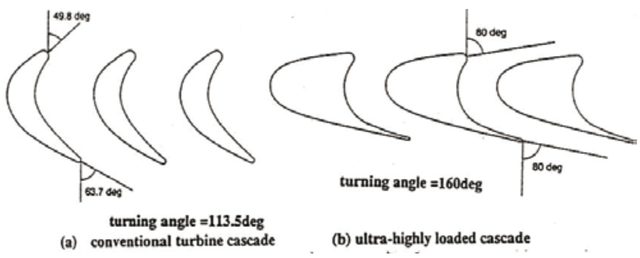

Fig. 2 *Comparison of rotor blade ring cascades:* ***a)*** *conventional and* ***b)*** *highly loaded, acc. [2].*

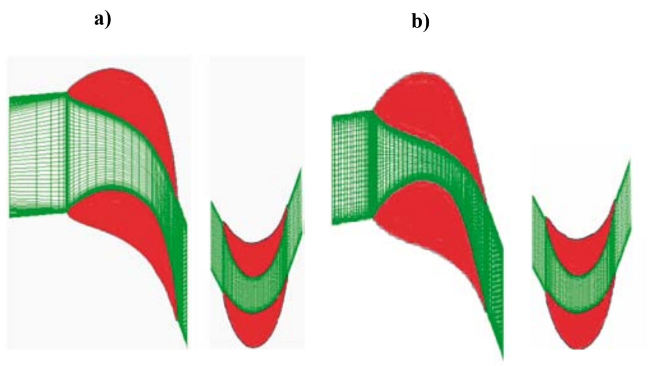

Fig.3 *Geometry of guide vane and rotor blade rings of the turbine stages:* ***a)*** *conventional and* ***b)*** *high-loaded*

that is resulted from a distinctly larger change of flow turning angle in blade rings. The so large changes of flow direction reaching even 160^0 are realized first of all due to change in suction (concave) side contours of profiles, as described in detail by Amos [3].

The larger flow turning angles result obviously in a greater loss of energy. In the tests performed on the new designed stages the turning angles were limited to the range of $130 \div 140^0$. In this case the guide vane ring profiles operate at outlet angles in the range of $10 \div 12^0$ and the rotor ring profiles – in the range of $16 \div 18^0$. Forms of the new profiles resulted from optimization of conventional impulse profiles elaborated in ALSTOM, Elblag, in co-operation with the von Karman's Institute, Brussels. Their geometry is schematically shown in Fig. 3.

The performed numerical analyses confirmed that the new profiles have been applicable to the assigned aims [4]. The relevant flow calculations were carried out by using the CFD FlowER program and 3D model

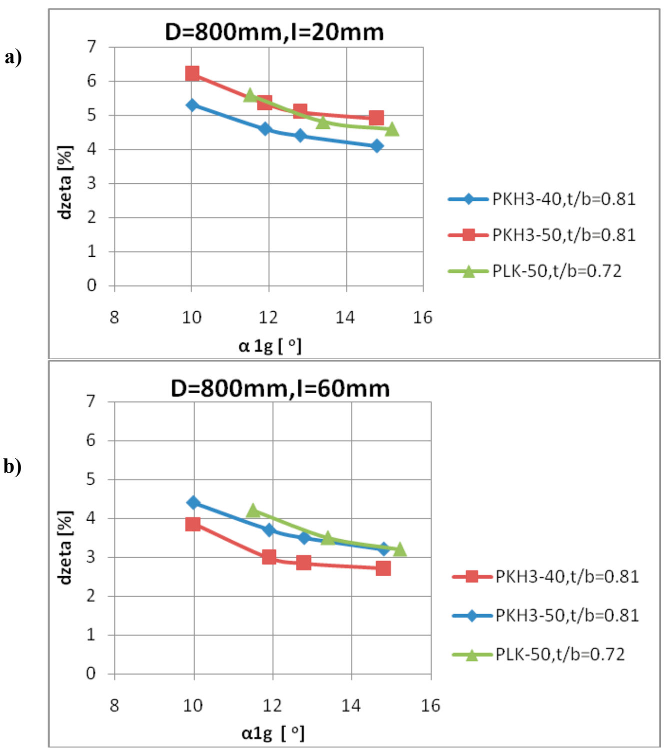

Fig.4. *Comparison of energy losses in function of the angle α1 for cylindrical blade ring cascades of;* ***a)*** *PKH3-40 profiles and* ***b)*** *PLK-50 profiles, acc. [4].*

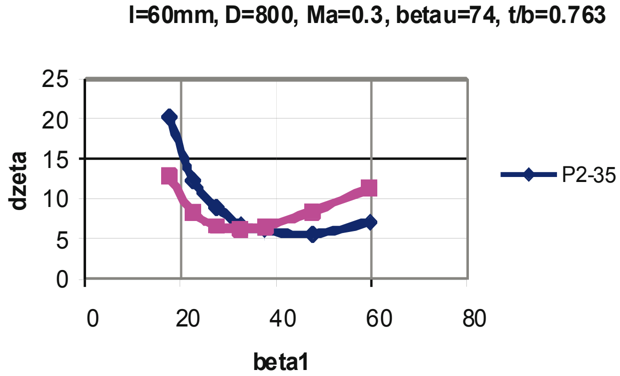

Fig.5. *Comparison of energy losses for P2-35 and P2V-35 profiles in function of the inlet angle beta 1, acc. [4]*

which takes into account viscosity of medium [5]. The calculations dealt with single ring of guide vane and rotor blades for geometry and operational conditions appropriate for operation in heat power turbine. It was observed that the losses within the cascades of guide vane blades of the new, PKH3 profiles, at the small values of the outlet angle $\alpha_1 \approx 11^0 \div 13^0$ and the velocities corresponding with higher loads, were lower by more than 1%, as compared with those recorded in the PLK typical impulse blade rings, Fig. 4.

In the PKH3-profile blade rings the lower level of flow losses resulted not only from the performed efficiency-orientated optimization of form of the profiles but also change of their geometry. In the new, „thicker" guide vane blade profiles it was possible to limit breadth of the ring at maintaining similar values of strength parameters. Such operation leads additionally to limitation of secondary and edge losses at the same

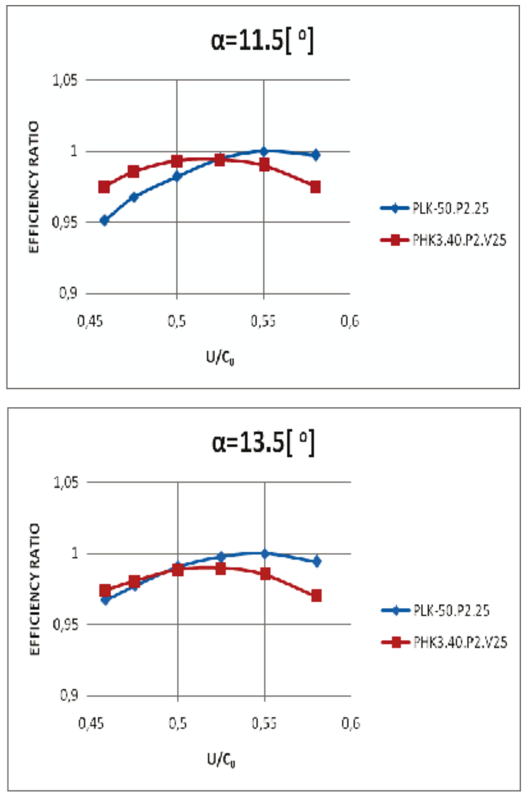

Fig.6 *Change of efficiency in function of loads for the group of three cylindrical stages with PKH-3/P2V and PLK-P2 profiles at different values of the inlet angle, α1, acc. [4].*

length of blades. It should be stressed that values of the optimum-efficient relative pitch of the new-profile rings exceeded 0.8, whereas values of the conventional relative pitch of guide vane rings were kept on the level of 0.7. It corresponds with a lower number of blades.

A little lower losses than those for P2 conventional profiles were also obtained for new rings of rotor blades of P2V profiles. The lower losses concerned flows at smaller inlet and outlet angles, characteristic for higher values of stage load. Comparison of the new and old profiles showed that the lowering of losses was especially distinct for values of the inlet angle $\beta_1 < 27^0$, that corresponded with values of the outlet angle $\beta_2 < 18^0$. The observed differences exceeded 2% in favour of the new profiles, however the level of minimum losses for greater angles corresponding with lower loads on conventional rings, was lower by 0.5%, (see Fig. 5).

The positive results of analysis of steam flow through rings of PKH3/P2V profiles encouraged to perform successive numerical tests. They dealt with investigations of turbine stages. In this case was searched for an optimum efficiency-orientated geometry of three cylindrical stages fitted with the new and conventional profiles, suitable for operational conditions of the first group of stages of the designed heat power turbine. It was revealed that the stages with PKH3 P2V profiles were more efficient than those with PLK P2 profiles at the same assumed diameter D_w = 800 mm, in the range of greater loads at the applied angle $\alpha_1 \sim 11.5^0$ (see Fig.6). In the diagram the loads are determined by the ratio u/co. The curve of efficiency change in the new stages appeared more „flat" as compared with the characteristics of typical impulse stages. The superior features are not observed at greater values of inlet angles, i.e. on the level of 13.5^0.

The relative pitch values of the calculated stages at which the highest efficiency was obtained, was determined on the level of 0.8 for the new, PKH3 guide vanes, and for the old, PLK ones on 0.7, at the similar pitch values for both rotors, on the level of 0.7. It resulted from the optimization process of mutual orientation of blades.

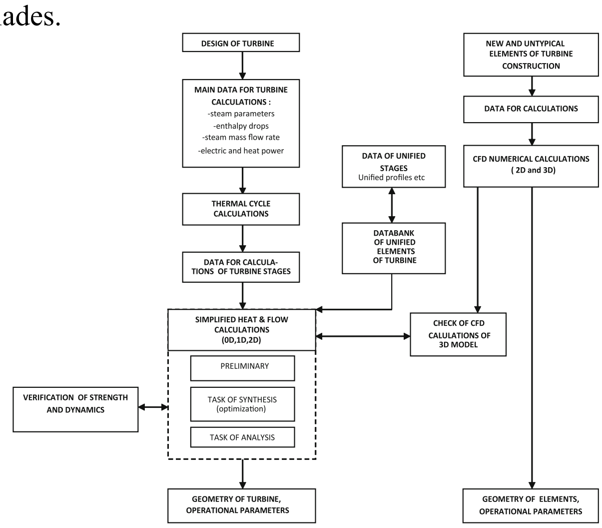

Fig.7 *CAD system for high-loaded turbine stages, acc. [6].*

Also in this case a favourable impact on efficiency was given by the smaller breadth assumed for the new guide vane profiles, that additionally made length of turbine flow passage, shorter. As above mentioned, the turbine stages with the new blade rings which permit higher loads to be applied, were practically installed in 7CK-65 turbine construction. Relevant design calculations were carried out with the use of the CAD system elaborated in Poland, whose schematic diagram is shown in Fig. 7 [6].

Data concerning thermodynamical parameters of the stages were determined from thermal cycle calculations of the turbine unit, with taking into account parameters of boiler, water cooling system, as well as the electric and heat power assumed for the unit. From preliminary calculations of the turbine (0D model) it has resulted that cylindrical stages could be used for three groups of stages. The following basic gabarites were determined for them: diameters, blade heights, as well as number of stages. The data were used as the basis for detail design calculations (synthesis) and check calculations (analysis) which were rperformed by means of simplified programs for 1D and 2D model. In the programs solutions of non-viscous-flow equations were corrected by means of coefficients obtained from CFD numerical calculations for blade rings with new profiles, in accordance with the concept presented in [7].

Efficiency optimization procedures built-in the programs, made it possible to determine passage geometry, as well as number of blades and their most favourable orientation in particular stage rings. In the calculations the producer's design standards concerning diameters, overlaps as well as solutions of sealing were taken into account [8].

The increasing of stage loads made it possible to reduce number of cylindrical stages in the turbine from 20 to 16. It was achieved at a high flow efficiency confirmed by numerical calculations of 3D model [9]. As obtained from the tests, the efficiency was -by a little more than 1% -different from the maximum value available in the case of conventional impulse stages.

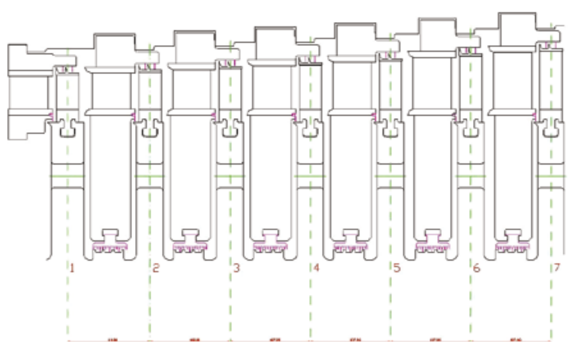

Fig.8 *Axial cross-section of 1st group blade system of 7CK-65 heat power turbine, acc. [8]*

In Fig. 8 the computer drawing of the flow system of 1st group cylindrical stages is presented.

The drawing was automatically generated on the basis of heat–and-flow calculations just after design verification of particular rings by means of strength and dynamics calculations [8]. As a result of the lowering of number of cylindrical stages it was possible to resign easily from an additional cylinder originally planned in the turbine. A photo of the turbine set and its axial cross-section covering also two steam -flow control stages intended for heating the extractions is shown in Fig. 9.

The calculations for the designed flow system of the turbine were verified experimentally in the heat-and-power station. To this end the routine guarantee tests of the entire turbine set were supplemented by thermal measurements in compliance with the concept given

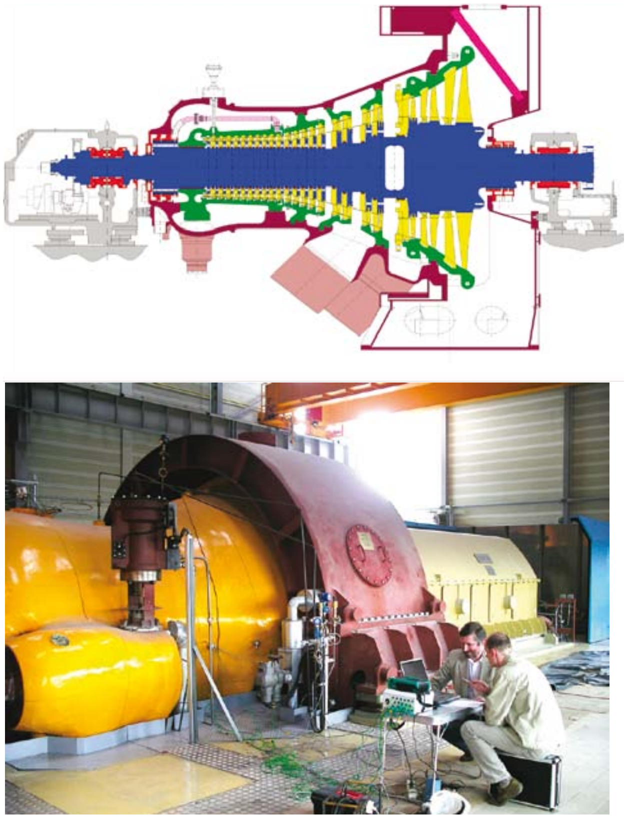

Fig.9 *Photo and axial cross-section of 7CK-65 steam turbine set, [4].*

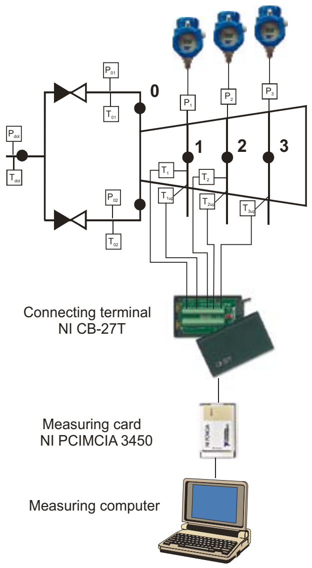

Fig.10 *Schematic diagram of the measuring system intended for the testing of the cylindrical stages of 7CK-65 turbine, acc.[11].*

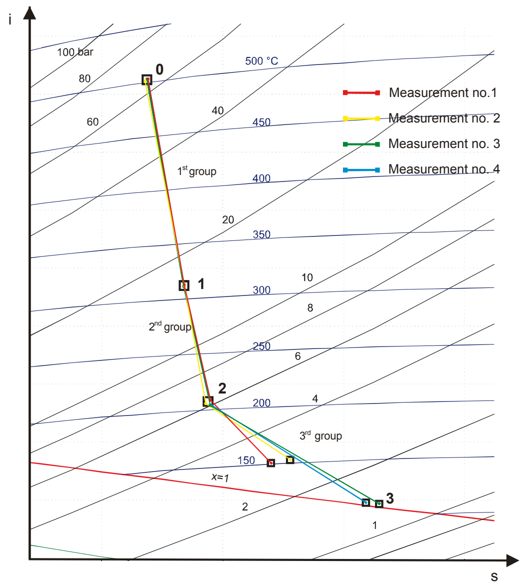

Fig.11 *Run of steam expansion line for 7 CK-65 turbine, acc. [10].*

in [10]. Schematic diagram of the measuring system intended for the testing of cylindrical stages is shown in Fig. 10, acc. [11].

The measured pressure and temperature values made it possible to determine steam enthalpy and expansion line in this part of the turbine (Fig. 11). The tests covered series of four measurements in various operational conditions. The performed analysis showed that the measurement errors in estimating efficiency of two first groups of the turbine's stages did not exceed 1%. The third group of the turbine's stages was excluded from the analysis because during the tests its operational parameters appeared greatly variable. It was caused by the fact that to this group the steam leaking from seals, largely changing the blade system parameters, was delivered.

Tab. 1 presents comparison of calculated and measured values for 1st groups of cylindrical stages in nominal conditions.

Tab. 1 *Comparison of calculated and measured values for 1st groups of cylindrical stages in nominal conditions*

	As measured	As calculated
Inlet pressure [MPa]	6.287	6.287
Inlet temperature [oC]	498	498
Outlet pressure [MPa]	1.553	1.553
Outlet temperature [oC]	310.1	307.5
Efficiency of the group of stages [-]	0.879	0.892
Mass flow rate [kg/s]	51.05	51.35

Only 1.5% differences in efficiency were obtained at a very high conformity of the calculated mass flow rate with that measured with the use of an area reducer. They were only a little smaller from measurement errors. As the analysis showed, the higher value of calculated efficiency resulted from the fact that non-stationary processes were not taken into account [9] during determination of the correction coefficients used in the design programs. The 7CK-65 one-cylinder turbine of 16 stages fitted with the new profiles has faultlessly operated for over seven years.

New solutions of stages before extraction

In construction of turbine stages to introduce clearance (gaps) between moving and motionless elements in order to ensure their co-operation is necessary. Steam fluxes flowing through such gaps do not transfer work to rotor, which results in loss of energy. The phenomenon is especially intensive in low-pressure parts of turbine where large gaps over unshrouded rotor blades are present and fast flowing steam makes medium outflow to regenerative extraction difficult. In the blade system whirl and stagnation zones are then formed, that was confirmed by probing measurements carried out on real turbines (see Fig. 12), acc. [12]. In order to eliminate the intensive fluxes it was proposed to introduce, to the stages placed before regenerative extraction, a ring which will direct the leaking steam to the heat exchanger directly, Fig.13. The relevant solution was claimed to Polish patent office [13] and sucessfully implemented in LP parts of 200 MW turbines with Bauman's stages [14].

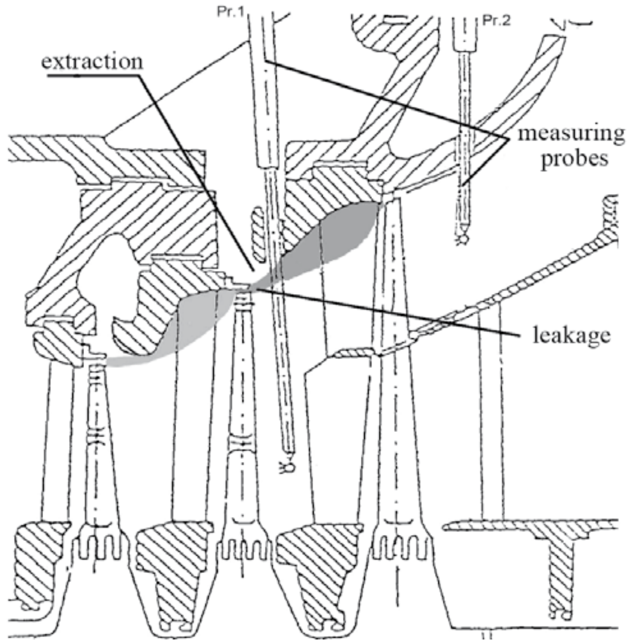

Fig. 12 *Low-pressure part of 200 MW turbine with measurement instrumentation. Separation zones in leakage area are shown in grey. wg [12].*

Advantages of application of the ring, with a view of turbine efficiency and operation, consisted in:

- elimination of whirl zones in the stage behind extraction both in meridional and circumferential plane;
- elimination of the mixing of two fluxes: of main flow and leaking flow;
- the making use of higher energy of leaking steam in the first, usually underheated, regenerative extraction;
- the lowering of moistness losses and errosion damage by effective separation of water drops to the extraction; its low cost as compared with other solutions.

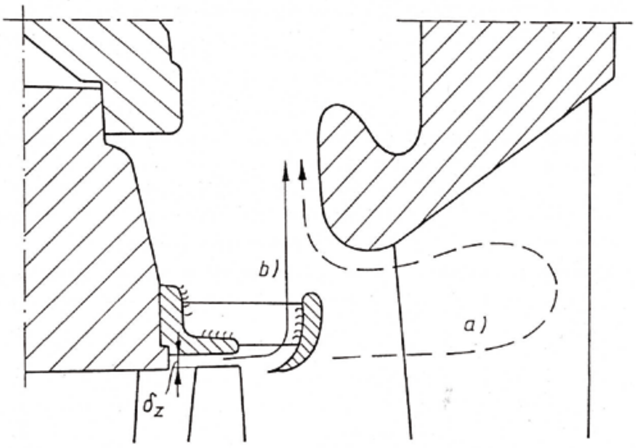

Fig.13 *New solution of turbine stage before extraction, acc. Patent no. P 160-805*

Both the performed theoretical investigations and measurements have demonstrated superiority of application of the new solution. Owing to it, shrouding seals -much more expensive and difficult in the case of realization of long blades -could be eliminated.

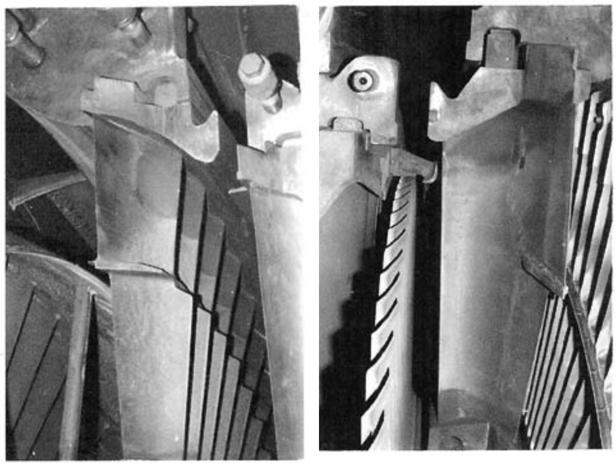

Fig.14 *Photos of 200MW turbine Bauman's stage in versions before (left) and after (right) its modernization, acc. [14].*

Photos of the turbine blade system in the versions before and after introduction of the ring are presented in Fig. 14. Salt sediments in the stagnation zones, eliminated after introduction of the ring, can be there observed.

In the years 1994÷2006 the solution was implemented in fourty 200 MW turbines in Poland and Ukraine. The average power increase of 600 kW/unit was obtained. The rings operated in the turbines faultlessly, that was confirmed by the observations carried out during operation [15].

The successful implementations have contributed to application of the considered solution also to the most modern design of LP parts fitted with ND 41A ALSTOM outlet, used for modernization of 200 MW turbines. In this case the decision on the application has been taken not so much due to profits in efficiency as to decrease faster erosion of inlet edge of blades of the last rotor stage, spread over 2/3 of their length (see Fig.15). This region was reached by water drops seperated in the next-to-last stage and intensively splashed by leakage flux. The recorded erosion damage are specially dangerous in unhardened areas shown in Fig. 15, acc. [16].

Fig.15 *Photos of inlet edge erosion of the blade of the last rotor stage after 40 000 h service, acc. [16].*

Fig. 16 presents the concept of introduction of the rings to the modernized LP part of 200 MW turbines, acc. [17].

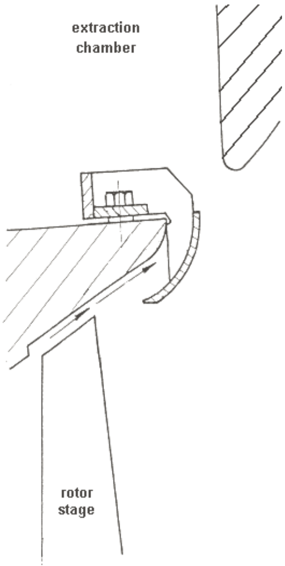

Fig.16 *Concept of introduction of the ring to the LP part of 200 MW turbine modernized in ALSTOM Works, acc. [17]*

After performing some additional analyses the patent was implemented to the power unit no.5 of Kozienice Electric Power Station. The analyses concerned detail heat-and-flow calculations in which values of its operational parameters before and after modernization were compared to each other. The CFD numerical calculations were carried out by means of the FlowER software [5]. For both the cases identical steam flow boundary conditions which resulted from the probing measurements carried out on the real power unit [18], were assumed. They covered distributions of static pressure, total pressure, temperature and steam flow angles before and behind stages, for a set value of steam mass flow rate. Description of the applied calculation methods can be found in [19].

The calculated total pressure distributions and streamlines prior assembling the rings in the diffuser between 3rd and 4th stage are presented in Fig. 17. The images distinctly show how the leakage blocks the steam flowing toward extraction.

In the performed analysis concerning the modernization the ring's form most favourable for efficiency was determined by using multi-variant numerical calculations. The following quantites were optimized: diameter of the pipe the ring was made of, its length, inlet edge orientation angle and its location with respect to rotor blade. The location was crucial for final choice as it had to guarantee that any intensive leakage is eliminated from the main steam flux at only minor disturbance of it. Some characteristic images of the distributions of flow parameters for the calculated variants, are given in Fig. 18.

Total pressure distribution and streamlines in the diffuser and extraction chamber for a selected variant are presented in Fig. 19. The leakage flows to the extraction chamber and after being braked it rather smoothly splits into the flow to the heat exchanger and the continued flow through the blade system. For the ring's position selected for nominal conditions, 3 kPa rise of pressure in the extraction chamber with respect to the initial version, was recorded. Also, the inflow to the guide vanes ring of the last stage has been favourably ordered, that is illustrated by the uniformly arranged stremlines. As calculated, the phenomenon resulted in the stage efficiency increase by over 1%. The shift of the ring inlet from the edge of the rotor blade should ensure effective sucking-off the water accummulated in this zone due to action of centrifugal forces, and to direct it into regenerative extraction and reduce erosion this way.

The performed calculations made it possible to assess forces acting on the ring, which were the basis for strength and dynamic verification. The recorded loads appeared

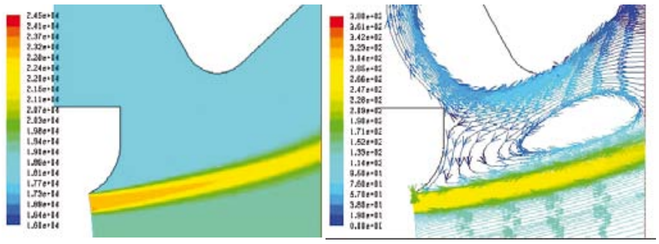

Fig. 17. *Total pressure distrbution (left) and streamlines(right) in the interstage diffuser, close to regenerative extraction, prior assembling the ring, acc. [18].*

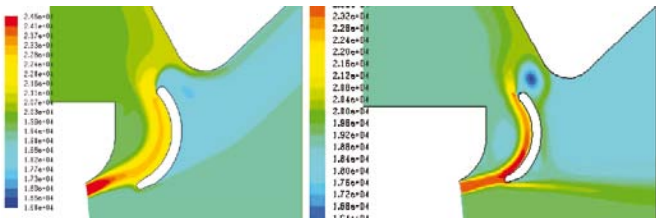

Fig. 18. *Distributions of steam flow total pressure close to extraction in turbine for selected calculation variants.*

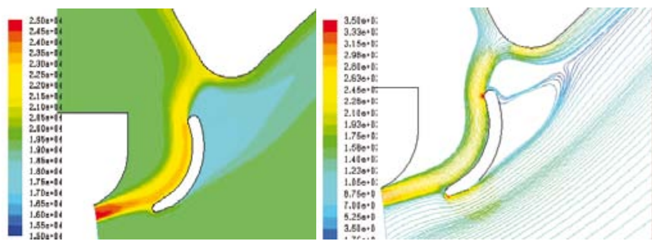

Fig.19. *Distrbutions of total pressure (left) and streamlines(right) in the interstage diffuser, close to regenerative extraction, after assembling the ring, acc. [18].*

rather small and danger of resonances effectively eliminated by proper choice of number of ribs fastening the rings to clamp [20], which was taken into account in the final design.

The significant changes in extraction chamber pressure and the last stage efficiency were used to assess energy profits to be obtained from application of the patent in question. They were assessed by calculations of thermal balance of the entire turbine set, performed with the use of the thoroughly tested DIAGAR software [21]. Main data for the above mentioned calculations were taken from measurements of thermodynamic parameters prepared on the basis of the guarantee tests [22].

Schematic diagram of thermal cycle of the power unit, i.e. 200 MW power unit no.5 at Kozienice Electric Power Station, is presented together with depicted measurement points in Fig. 20.

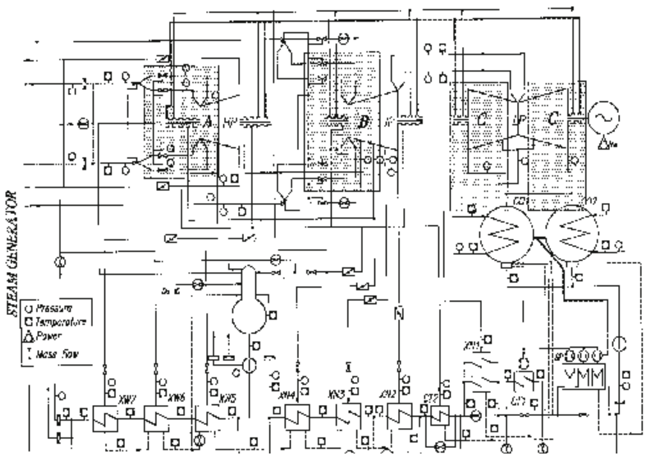

Fig. 20. *Schematic diagram of heat-and-flow cycle of the power unit, i.e. 200 MW power unit No.5 at Kozienice Electric Power Station, with depicted measurement points, acc. [22].*

Changes of output power and specific heat consumption in the cycle after implementation of the patent to the modernized turbine fitted with ND41 outlet in function of output power and condenser pressure, are shown in Fig. 21.

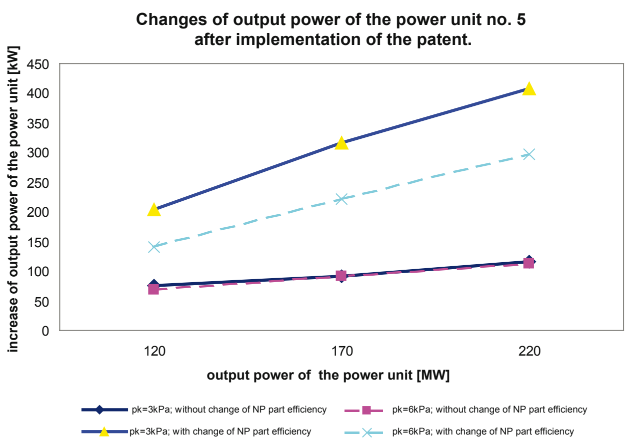

Fig.21 *Changes of output power and specific heat consumption for various loads of the turbine set after implementation of the patent in question, acc. [19].*

As can be observed in the figure, within the turbine load range of 120÷220 MW and the condenser pressure range of 3÷6 kPa the decrease of specific heat consumption amounts to 11÷16 kJ/kWh, which results in the output power increase of 200÷400 kW. The equivalent increase of the turbine-set efficiency is in the range of 0.1÷0.2% which can not be measured during routine guarantee tests.

The efficiency rise effects can be only determined indirectly during measurements after modernization. It concerns the easily measurable rise of feed-water temperature in the first heat exchanger, by 3^0C. Measurements of distribution of thermodynamical parameters can be used in interstage spaces, which can be realized by the probing method only. Such measurements are planned to be performed soon on the turbine set no. 5 at Kozienice Electric Power Station (see Fig. 22). Within the scope of the measurements control of erosion damage growth in the last stage rotor blades as well as inlet area of the rings, is provided for.

Fig.22. *The ring placed inside 225 MW turbine*

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