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CONCEPT OF K-300 SERIES STEAM TURBINE FLOW PART MODERNIZATION FOR TRANSITION TO OPERATION WITH ULTRA-SUPERCRITICAL STEAM PARAMETERS

Roman A. Rusanov
roman_rusanov@ipmach.kharkov.ua
ORCID: 0000-0003-2930-2574

Anatolii Pidhornyi Institute of Power Machines and Systems of NAS of Ukraine,
2/10, Komunalnykiv str., Kharkiv, 61046, Ukraine

A concept of the K-300 series steam turbine flow part for transition to operation with ultra-supercritical initial steam parameters is described in the paper. A loop scheme with two-tier blades of the steam turbine flow part has been proposed for the first time in the world. The main turbine parameters, such as pressure and temperature at the inlet to the high-pressure cylinder (HPC), temperature of the intermediate superheat, temperature and mass flow rate at the outlet from the low-pressure cylinder, were selected. The turbine is designed to operate with initial parameters of fresh steam at a pressure of 35 MPa and a temperature of 700 °C with intermediate steam superheating to 700 °C. The flow part was divided into cylinders with a preliminary estimate of the number of HPC and intermediate-pressure cylinder (IPC) stages and determination of their axial dimensions. The feasibility of using a loop scheme with two-tier blades in HPC and IPC flow parts was substantiated, and thermal drop in stages were determined. The main geometric characteristics of the HPC and IPC stages were determined taking into account the loop scheme and two-tier blades. A three-dimensional model was developed and the flow in the turbine flow parts was calculated. The obtained results show a high internal efficiency of the new HPC and IPC flow parts of 94.18% and 94.5%, respectively. This will increase the efficiency of the power plant up to 49.2% and provide an increase in capacity by 80.64 MW.

Keywords*: steam turbine, ultra-supercritical steam parameters, gas-dynamic efficiency, mathematical modeling, flow part.*

Introduction

Lately, the global energy industry has a steady increase in the efficiency of steam turbine power units at thermal power plants (TPP). This is achieved by improving operating cycles, developing new equipment, and increasing the initial parameters of the steam [1]. The prerequisites for increasing the technical efficiency of TPP power units are: progress in the development of new materials, improvement of power equipment and the desire to reduce the negative impact on the environment, including control of CO_2 emissions. The efficiency of the power unit can be increased from 37 to 42% by improving the turbine [2] and boiler [3] equipment, optimizing the operating cycle [4]. Further increase in energy efficiency to the level of 47–55% is achieved by increasing the initial steam parameters [5].

Achieving higher initial steam parameters has always been a top priority for turbine manufacturers, as the higher the initial steam parameters are, the better turbine efficiency is. Today, the benchmark that world leaders in turbine construction strive for is ultra-supercritical steam parameters (up to 35 MPa and 750 °C) (Fig. 1) [6, 7].

The world leaders in the development and production of steam turbines are the following companies: Siemens [8], General Electric [9], Mitsubishi Power [10], JSC "Ukrainian Energy Machines" [11], and others.

General Electric is developing turbines with a capacity of 600–1100 MW with initial temperature parameters up to 670 °C and pressure up to 330 bars [12], which allows reducing CO_2 emissions into the atmosphere by 26%. Siemens plans to deploy its 660 MW ultra-supercritical steam turbine technology with the highest known steam initial parameters in the world (325 bars and 670 °C) [13]. Mitsubishi Power is working on creating 1000 MW turbines with ultra-supercritical steam parameters with initial steam parameters up to 630 °C and 330 bars [14].

As a result of massive attacks by the aggressor country on Ukraine's energy infrastructure, most TPPs suffered significant damage [15]. In addition, most of the TPP units with a capacity of 200+ and 300+ MW have exhausted their installed and extended resources [16].During repairs and/or restoration, it is possible to convert these power units to ultra-supercritical initial steam parameters. This approach will give good prospects for initial costs, as it will allow to preserve some of the equipment unchanged [17]. A concept of modernization of the K-330-23.5 series turbine with the transition to ultra- supercritical initial parameters, in which it is possible to preserve the existing intermediate (IPC)and low-pressure cylinders (LPC) unchanged, is described in the paper.

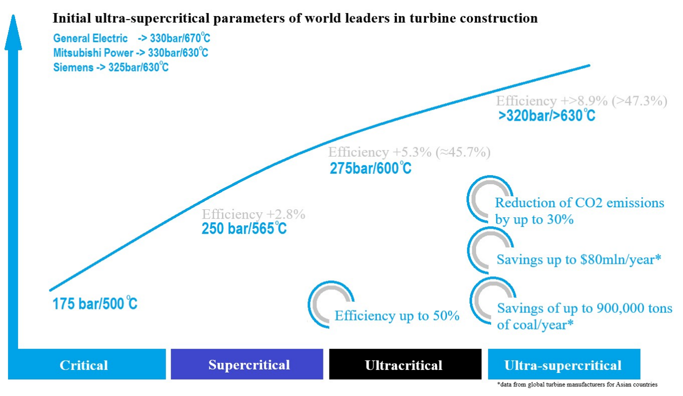

Fig. 1. Types of initial steam parameters

Mathematical model

The design and calculations of an axial turbine flow part are carried out according to the algorithm implemented in the IPMFlow software package using methods and models of various levels of complexity, from one-dimensional methods for selecting the main characteristics of the turbine stage [18], methods for analytical profiling of axial stages [19], to the methods for calculating three-dimensional viscous flows in the turbine flow parts [20].

Determination of the main thermodynamic characteristics of the new flow part stages

The development of a power unit for ultra-supercritical initial steam parameters is a difficult, complex task, in which, perhaps, the most problematic component is the boiler. An important component – the turbine – is also considered in this paper. An option of the K300 series turbine, namely K-330-23.5, the mid-meridional section of which is shown in Fig. 2, was chosen as the basis.

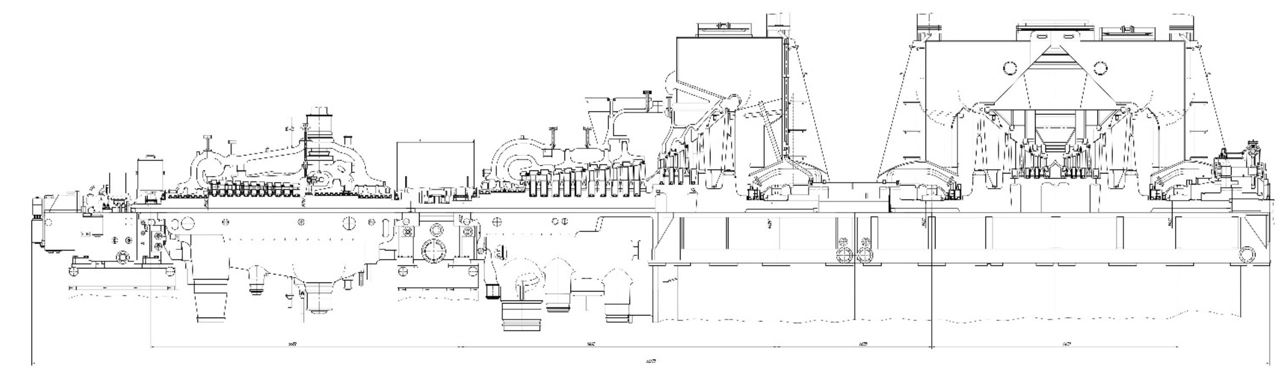

Fig. 2. Mid-meridional section of the K-330-23.5 turbine flow part

This option has the following initial parameters: temperature 560 °C and pressure 23.54 MPa, intermediate superheat temperature is also 560 °C, and separating pressure is 3.97 MPa. To create a new flow part, the initial temperature was assumed to be 700 °C, the pressure is 35 MPa, and the intermediate superheat temperature is also taken to be 700 °C. The distribution pressure must be calculated based on reaching a point on the i-s diagram that corresponds to the parameters at the inlet to the existing IPC (Fig. 3). Under such conditions, the flow part of the existing turbine IPC and LPC can be left unchanged.

After point number 3, the turbine remains unchanged. In the new turbine, the thermal drop that needs to be worked on is much greater, in fact, a completely new high-pressure cylinder (HPC) is required, as well as additional stages at the beginning of the IPC. The intermediate superheat in the new turbine shifts to the higher-pressure region.

At the first step, using one-dimensional methods implemented in the IPMFlow software package, calculations were carried out to determine the number of stages, thermal drop and other basic gas-dynamic and geometric characteristics of the flow part blade rows. The thermal drop for each stage was chosen as the maximum possible with the optimal value of u/c=0.7 (reactive type stage) and a degree of reactivity of no more than 0.5 [21]. The predetermined value of the thermal drops in stages, and the division into cylinders, is given in Table 1. The constants of the Tamman equation of state for further determination of the main geometric characteristics are also given in the Table 1.

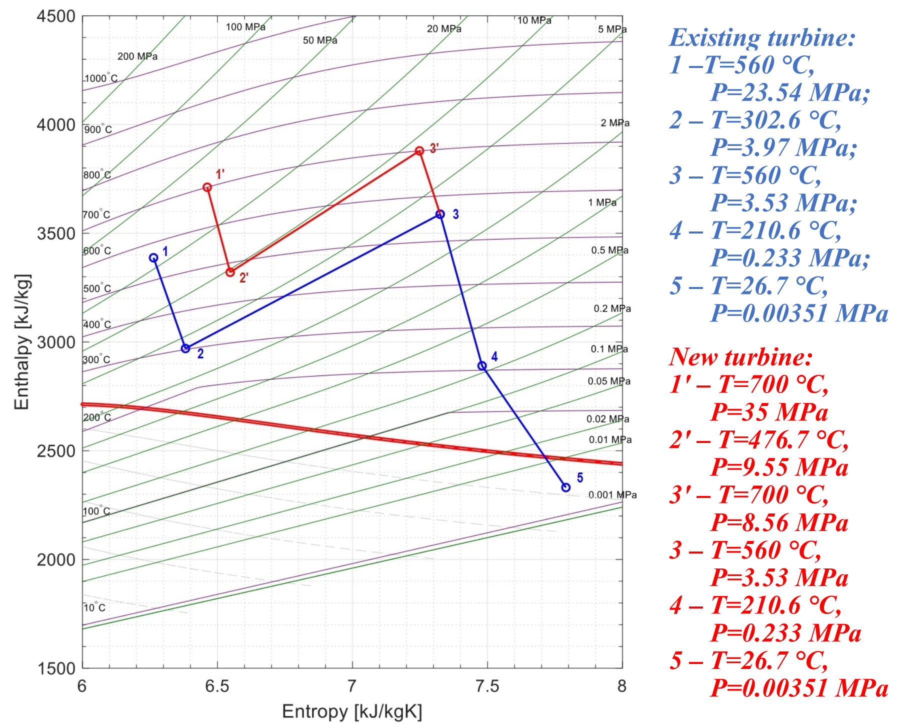

Fig. 3. i-s diagrams of operation of the existing and new turbine

Table 1. Thermal drops and constants of the Tamman equation

Stage no.	H*, j/kg	ρ*, kg/m^3	H_2^{iz}, J/kg	ρ_2^{iz}, kg/m^3	P_0, Pa	γ	R, J/(mol·K)
HPC							
1	3711550.40	86.78647	3681694.90	81.96450	483331.92	1.28797	420.1386
2	3683784.78	81.86918	3654010.88	77.24552	348012.02	1.28796	418.5885
3	3656095.05	77.15418	3626401.62	72.72624	238584.62	1.28802	417.2607
4	3628480.16	72.63879	3598866.51	68.40351	150862.03	1.28814	416.1366
5	3600939.47	68.31987	3571404.91	64.27373	81488.44	1.28831	415.2027
6	3573472.33	64.19382	3544016.21	60.33288	27715.82	1.28853	414.4493
7	3546078.14	60.25660	3516699.72	56.57657	-13444.63	1.28880	413.8563
8	3518756.22	56.50384	3489454.95	53.00013	-43985.78	1.28912	413.4154
9	3491506.03	52.93084	3466121.65	50.02787	-64651.71	1.28945	413.1407
10	3467898.56	49.97024	3442376.64	47.17236	-78397.53	1.28979	412.9730
11	3444163.18	47.11679	3418502.53	44.42324	-87580.50	1.29017	412.8992
12	3420298.78	44.36969	3394498.16	41.77951	-93235.92	1.29057	412.9069
13	3396304.20	41.72797	3370362.33	39.24005	-95718.66	1.29101	412.9993
14	3372178.27	39.19050	3346093.86	36.80356	-96028.89	1.29146	413.1585
15	3347919.77	36.75598	3321691.48	34.46863	-94095.77	1.29194	413.3982
16	3323527.46	34.42298	3297012.81	32.22224	-90716.25	1.29244	413.7017
Additional stages of the IPC							
1	3876222.4	20.53382	3834189.42	18.87431	-11223.23	1.26467	448.8318
2	3834189.42	18.87431	3792789.16	17.34239	-12967.11	1.26611	448.8104
3	3792789.16	17.34239	3752016.89	15.92884	-13821.98	1.26759	448.8318
4	3752016.89	15.92884	3711867.90	14.62503	-14009.67	1.26911	448.8918
5	3711867.90	14.62503	3672337.49	13.42289	-13769.40	1.27065	448.9817
6	3672337.49	13.42289	3646245.61	12.67223	-13238.16	1.27193	449.0974
7	3646245.61	12.67223	3619567.42	11.93838	-12745.23	1.27301	449.1890
8	3619567.42	11.93838	3592294.42	11.22226	-12160.11	1.27414	449.2943
9	3592294.42	11.22226	3564417.93	10.52472	-11537.34	1.27531	449.4090
10	3564417.93	10.52472	3535929.05	9.84660	-10795.59	1.27653	449.5440

Where: H^* – enthalpy at the inlet to the stage; ρ^* – density at the inlet to the stage; H_2^{iz} – isentropic enthalpy at the outlet from the stage; ρ_2^{iz} – isentropic density at the outlet from the stage; P_0 – Tamman equation of state constant; γ – adiabatic index; R – universal gas constant.

As can be seen from Table 1, a larger number of stages is required to operate a larger thermal drop, and taking into account the fact that for greater efficiency it is desirable to switch to reactive-type blades instead of the active-type ones, which are traditional for Kharkiv turbines, the number of stages also increases.

Problem statement and ways to solve it

When using ultra-supercritical initial steam parameters, a number of non-trivial problems arise, the main ones of which are the following:

– the first one is that to operate a larger thermal drop, a significantly larger number of stages is required, which is difficult to provide in the existing axial dimensions. Preliminary estimates showed that the new turbine, using reactive-type flow parts, will have 16 HPC stages and 10 additional IPC stages. That is, there will be 26 stages instead of 12;

– the second one is that at such high temperatures and pressures, problems with the strength of structural elements arise;

– and the third one, less complex but important problem, is that when using reactive-type stages, axial loads increase, which must be compensated.

To solve these problems, for the first time in the world, a fundamentally new solution, which has never been used before for turbines, both steam and gas ones, has been proposed in the paper. This is the so-called loop-type flow part with two-tiered blades. A utility model patent has been issued for this solution [22]. In this option, the steam will pass as shown in Fig. 4, half of the stages will be on top and half on bottom.

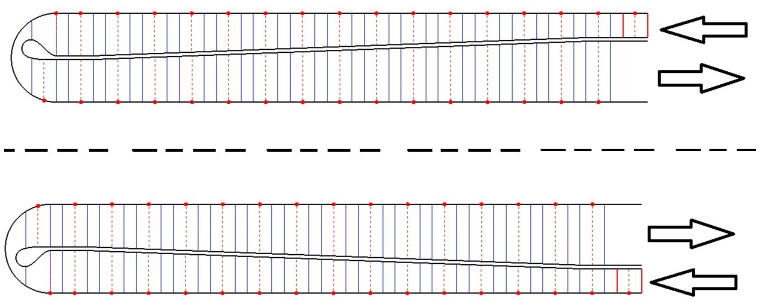

Fig. 4. Mid-meridional section of the loop-type flow part with two-tiered blades

This approach offers good prospects for solving these problems:

– first, the axial size for the same number of stages is almost halved;

– second, to solve the problems of blade strength, it is not necessary to use expensive materials; with a loop-type scheme, internal cooling can be organized when the hotter upper blades are cooled by colder steam from the lower part, while the heat is not lost, but preserved and is used in the thermodynamic cycle;

– third, due to the fact that the flow from above and below moves in opposite directions, there is partial compensation of axial forces.

Thus, the loop-type scheme actually makes it possible to solve the problems that arise when switching to ultra-supercritical initial steam parameters.

Development of options of the HPC and IPC steam turbine flow parts to operate with ultra-supercritical steam parameters

Using one-dimensional methods implemented in the IPMFlow software package, as well as data from Table 1, the main geometric characteristics of the HPC and additional stages of IPC flow parts were determined. The number of blades is selected so that their number at the lower part of the turbine corresponds to the number of blades in the corresponding opposite stage at the top of the loop-type flow part. The main geometric parameters of the flow part of the HPC and additional stages of IPC are given in Table 2.

Where α, β – flow angles in absolute and relative motion; index 0 – at the stator inlet; 1 – at the stator outlet; 2 – at the rotor outlet.

The final designing of the flow part was carried out using the IPMFlow software package. A mid-meridional section and a 3D model of the new HPC option is shown in Fig. 5, and additional IPC stages are shown in Fig. 6.

Table 2. Main geometric parameters of the flow part

Stage no.	Stator blades no.	Rotor blades no.	D_{mid}, mm	L_{stator}, mm	L_{rotor}, mm	α_0, deg	α_1, deg	β_1, deg	β_2, deg	α_2, deg
HPC										
1	120	120	1103	20.00	20.00	0	78.00	71.52	-72.68	-7.90
2	120	120	1101	20.37	21.50	0	74.64	68.88	-69.88	-0.10
3	120	120	1099	22.62	23.75	0	72.40	66.47	-67.64	-8.38
4	120	120	1097	24.87	26.00	0	77.44	72.77	-73.80	-5.82
5	120	120	1094	27.12	28.25	0	77.00	69.69	-72.95	-9.14
6	120	120	1092	29.37	30.50	0	77.00	70.05	-72.95	-9.98
7	120	120	1090	31.62	32.75	0	77.00	70.78	-72.41	-9.88
8	120	120	1088	33.87	35.00	0	76.73	70.43	-72.14	-5.80
9	120	120	1012	35.00	35.00	0	75.92	69.68	-71.33	-5.87
10	120	120	1014	36.13	37.25	0	74.03	68.25	-69.17	-5.392
11	120	120	1016	38.38	39.50	0	73.26	67.94	-68.07	-5.49
12	120	120	1018	40.63	41.75	0	75.13	69.27	-69.52	-5.07
13	120	120	1020	42.88	44.00	0	77.00	64.64	-67.82	-8.71
14	120	120	1022	45.13	46.25	0	78.50	61.17	-65.94	4.29
15	120	120	1024	47.38	48.45	0	78.00	63.35	-66.30	3.80
16	120	120	1027	49.63	50.00	0	78.00	64.86	-69.72	6.08
Additional stages of the IPC										
1	70	70	1231	67.50	67.50	0	78.00	61.64	-65.52	1.57
2	70	70	1220	69.04	74.67	0	78.00	64.77	-67.86	-3.32
3	70	70	1210	80.31	85.95	0	78.00	63.31	-67.86	7.18
4	70	70	1200	91.59	97.22	0	77.00	63.56	-67.86	-17.69
5	70	70	1190	102.86	108.50	0	77.00	63.76	-67.08	-13.43
6	70	70	966	108.50	108.50	0	77.00	62.15	-65.52	3.48
7	70	70	976	114.14	119.78	0	77.00	61.45	-66.30	7.49
8	70	70	986	125.41	131.05	0	78.00	61.58	-67.25	2.15
9	70	70	996	136.69	142.33	0	78.00	62.73	-67.50	5.18
10	70	70	1007	147.96	149.50	0	78.00	62.01	-67.42	5.94

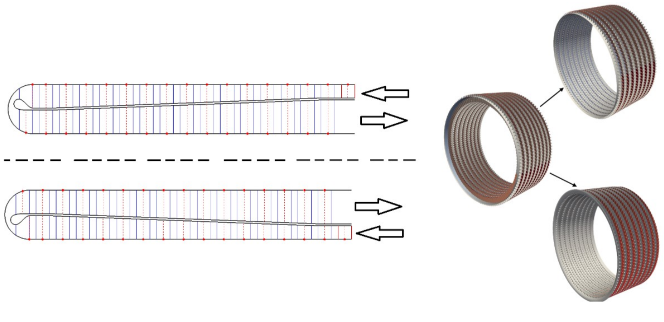

Fig. 5. Mid-meridional section of the new HPC option

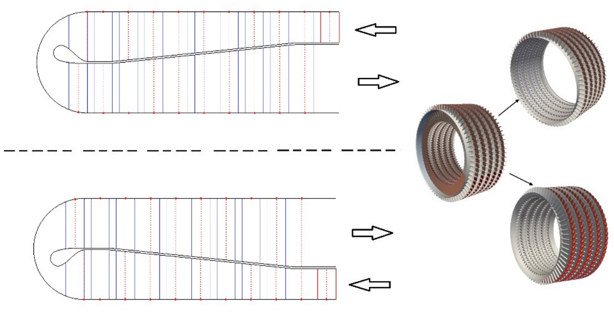

Fig. 6. Mid-meridional section of additional IPC stages

The new HPC option contains 16 reactive-type stages, as well as the additional 10 reactive-type IPC stages. Blades of all stages are designed with a constant height cross-section. The loop scheme allows to reduce the axial size for the same number of stages by almost half. From Fig. 7 it is clear that the new turbine will fit into the existing axial dimensions.

Spatial calculations of the steam flow in the HPC flow part were carried out on difference mesh with a total number of cells of about 16.5 million. A viscous turbulent flow model, based on the numerical integration of the averaged system of Navier-Stokes equations, was used [23]. Turbulent phenomena were taken into account using the two-parameter differential SST Menter turbulence model [24]. The IAPWS-95 equation of state [25] is used to describe the thermodynamic dependencies.

The distribution of static pressure on the blades surfaces along the middle cross-section is shown in Fig. 8, and the isolines of static pressure along the flow part middle cross-section are shown in Fig. 9.

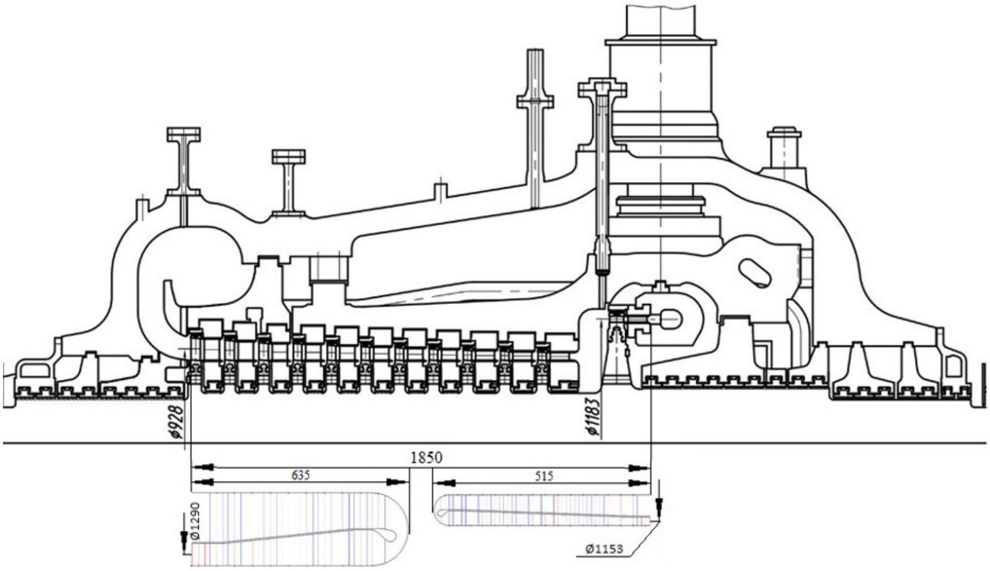

Fig. 7. Comparison of the geometric dimensions of the existing HPC with the new HPC and IPC (the proportions in the figure are not maintained)

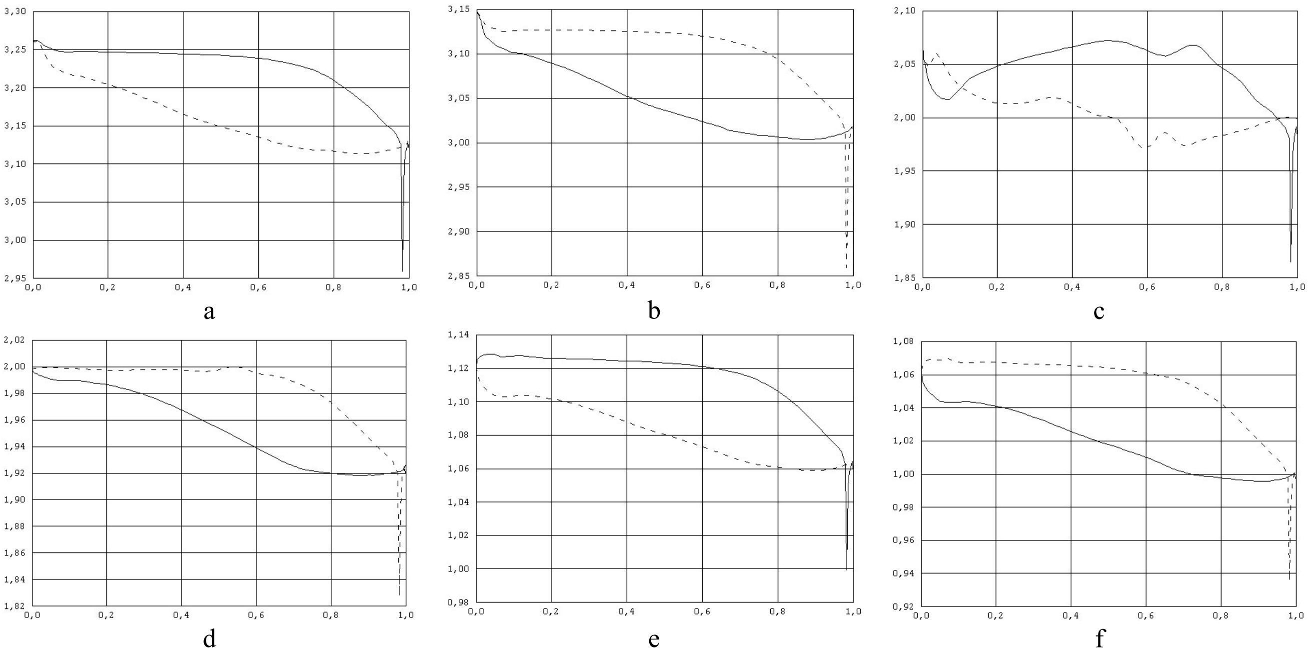

Fig. 8. Static pressure distribution on the blade surfaces along the middle cross-section:
a – 1 stage stator; b – 1 stage rotor; c – 9 stage stator; d – 9 stage rotor; e – 16 stage stator; f – 16 stage rotor

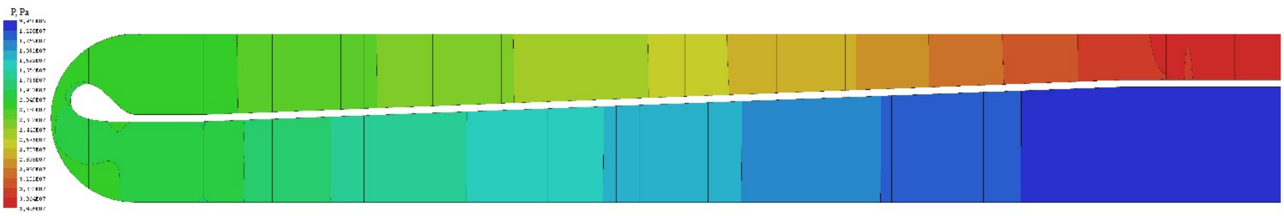

Fig. 9. Isolines of static pressure in the HPC mid-meridional section

The given results show that the flow pattern is very favorable, the pressure distributions are monotonous, except for the flow reversal zone (at the inlet to the 9th stage stator), there are no flow separations. Calculations were carried out taking into account regenerative steam extraction, but without taking into account leakage and radial clearances. Regenerative steam extraction is located in front of the 15th stage stator with a steam extraction of 21.13 kg/s. Despite the presence of flow separations at the flow reversal, a fairly high efficiency of the flow part was achieved – 94.18%.

Spatial calculations of the steam flow in the IPC flow part were carried out on difference mesh with a total number of cells of about 10.5 million. To study the steam flow in the IPC, the same mathematical model was used as for the HPC flow part.

The distribution of static pressure on the blades surfaces along the middle cross-section is shown in Fig. 10, and the isolines of static pressure along the flow part middle cross-section are shown in Fig. 11.

The given results show that the flow pattern is very favorable, the pressure distributions are monotonous, except for the flow reversal zone (at the inlet to the 6th stage stator), there are no flow separations. Calculations were carried out without taking into account leakage, radial clearances and regenerative steam extraction. Despite the presence of flow separations at the flow reversal, a fairly high efficiency of the flow part was achieved – 94,5%. This allowed to increase the efficiency of the entire power unit to 49.2%, i.e. 5.2% more than the existing design of the K-330 series power unit, and to ensure an increase in capacity by 80.64 MW.

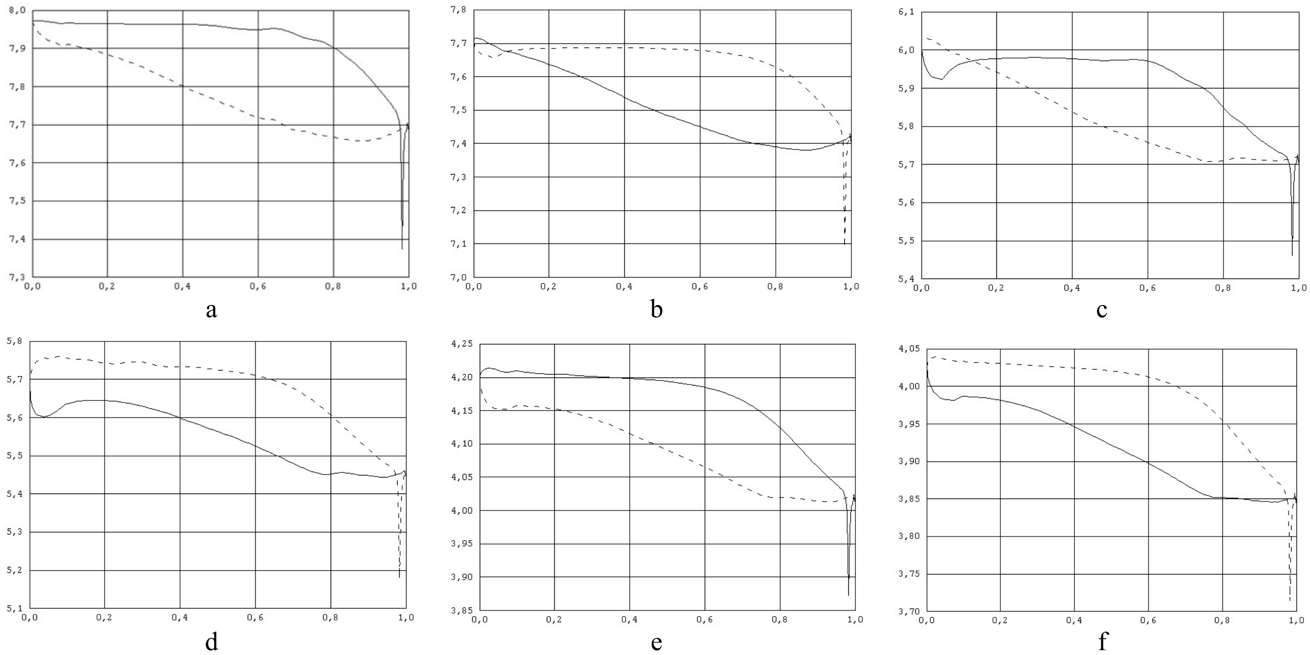

Fig. 10. Static pressure distribution on the blade surfaces along the middle cross-section:
a – 1 stage stator; b – 1 stage rotor; c – 6 stage stator; d – 6 stage rotor; e – 10 stage stator; f – 10 stage rotor

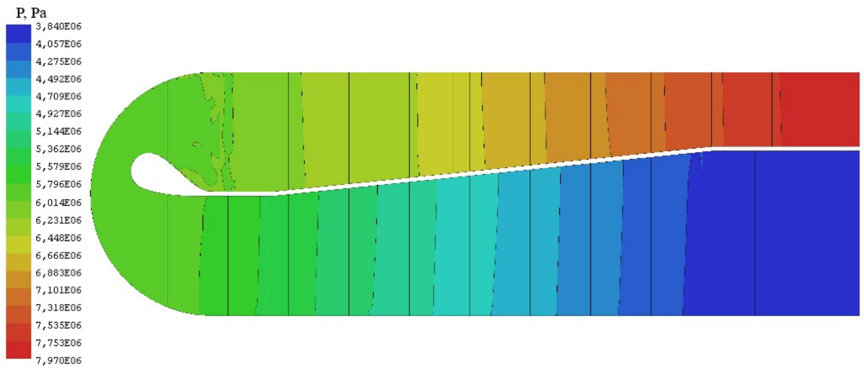

Fig. 11. Isolines of static pressure in the additional IPC mid-meridional section

At the next stage, calculations were made on the strength of the created flow parts. The locations of the greatest stresses were identified. They are located at the points where the blades are attached to the contours. The values of these stresses are permissible, but can be reduced by increasing the radius of the fillets. The results of strength calculations are given in the paper [26].

Due to the high temperature load of the first stages, it was decided to implement internal cooling of the blades by extracting colder steam from the "lower" blade. Numerical simulation of the temperature state of the HPC and IPC first and last stages rotor blades of a loop-type turbine has been carried out. It was determined that when using 2% of the steam mass flow rate for cooling, a decrease in the average blade surface temperature by 62.2 ºC for the HPC and by 40.4 ºC for the IPC is achieved, which is quite sufficient. The results of calculations of the temperature state of the blades are given in the paper [26]. A utility model patent was obtained for this solution [27].

Conclusions

For the first time, a loop-type flow part scheme has been proposed for steam turbines, which has a number of advantages and allows creating a new, highly efficient turbine for ultra-supercritical initial steam parameters. This approach offers good prospects for solving a number of problems. The axial size for the same number of stages is reduced almost by half, which will allow the new flow parts to be located into the axial dimensions of the existing one. To solve the problems of blade strength, it is not necessary to use expensive materials; with a loop-type scheme, internal cooling can be organized, when the hotter upper blades are cooled by colder steam from the lower part, while the heat is not lost, but remains and is used in the thermodynamic cycle. Also, due to the fact that the flow from above and below moves in opposite directions, there is partial compensation of axial forces. Highly efficient flow parts, which have a high efficiency of 94.18% for HPC and 94.5% for additional IPC, have been developed. The overall efficiency of the power unit increased to 49.2% (an increase of 5.2%), and the increase in capacity amounted to 80.64 MW.

Gratitude

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Концепція модернізації проточної частини парової турбіни серії К-300 для переходу на ультра-суперкритичні параметри пари

Р. А. Русанов

Інститут енергетичних машин і систем ім. А. М. Підгорного НАН України,
61046, Україна, м. Харків, вул. Комунальників, 2/10

У статті запропоновано концепцію модернізації проточної частини парової турбіни серії К-300 для переходу на ультра-суперкритичні параметри. Вперше у світі запропоновано/ представлено петльову схему з двоярусними лопатками проточної частини парової турбіни. Проведено вибір основних параметрів турбіни, таких, як тиск і температура на вході до циліндра високого тиску (ЦВТ), температура проміжного перегріву, температура і масова витрата на виході з циліндра низького тиску. Турбіна розрахована для роботи з початковими параметрами свіжої пари тиском 35 МПа і температурою 700 °C з проміжним перегріванням пари до 700°C. Здійснена розбивка проточної частини на циліндри з попередньою оцінкою кількості ступенів ЦВТ і циліндра середнього тиску (ЦСТ) і визначенням їх осьових розмірів. Обґрунтована доцільність використання петльової схеми із застосуванням двоярусних лопаток у ЦВТ і ЦСТ, проведена розбивка теплових перепадів по ступенях. Визначено основні геометричні характеристики ступенів ЦВТ і ЦСТ з урахуванням петльової схеми і двоярусних лопаток. Розроблено тривимірну модель і здійснено розрахунок течії проточних частин турбін. Отримані результати показують високий внутрішній ККД нових проточних частин ЦВТ і ЦСТ – 94,18% та 94,5 % відповідно, що дозволить збільшити ККД енергоустановки до 49,2% та забезпечить приріст потужності на 80,64 МВт.

***Ключові слова**: парова турбіна, ультра-суперкритичні параметри пари, газодинамічна ефективність, математичне моделювання, проточна частина.*

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