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Evaluation of the Thermal Condition of Marine Gas Turbine Components via Telemetric Measurements

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ABSTRACT

This paper focuses on assessing the thermal state of a marine gas turbine plant (MGTP), with particular attention being given to the measurement of turbine rotor component temperatures using a specially developed telemetric measurement system (TMS). The study was conducted within the framework of research aimed at evaluating the effects of reduced relative air temperature, which is supplied by the high-pressure compressor (HPC) through nozzle guide vanes (NGV) and a twisting grid (TG) inside the rotor blades (RB) of a high-pressure turbine (HPT), on the thermal state of the HPT rotor. This temperature reduction is achieved via a pre-swirl technique, in which the cooling air is directed toward rotor rotation. The results obtained demonstrate new opportunities for monitoring the thermal condition of turbine components during the modernisation of MGTPs. This approach contributes to improving the overall efficiency of the system by enabling an increase in the gas flow temperature at the inlet to the HPT's first stage.

Keywords: cooling system, telemetric measurement system, guide vane, high-pressure turbine, marine gas turbine plant

NOMENCLATURE AND UNITS

Abbreviations		
CC	Combustion chamber	
GTE	Gas turbine engine	
HPC	High-pressure compressor	
HPT	High-pressure turbine	
MCTC	Microcable thermocouple	
MGTP	Marine gas turbine plant	
NGV	Nozzle guide vane	
RB	Rotor blade	
TBC	Thermal barrier coatings	
TG	Twisting grid	
TMS	Telemetric measurement system	
Symbols and units		
C	velocity of gas in absolute motion	m/s
C_u	circumferential component of velocity in absolute motion	m/s
k	isentropic index	
P	Pressure	Pa
P_{in}	inlet pressure	Pa
~~Pout~~	outlet pressure	Pa
R	universal gas constant	J/(kg · K)
T	static temperature	K
T^*	total temperatures in absolute motion	K
T^*_w	total temperatures in relative motion	K
ΔT	reduction in air temperature	K
U	circumferential velocity of RB	m/s
W	velocity of gas in relative motion	m/s
W_u	circumferential component of velocity in relative motion	m/s
C_p	heat capacity	J/(kg · K)
ξ	hydraulic resistance coefficient of the sub-twisting grid	
P_{avr}	average flow density	kg/m³

INTRODUCTION

Enhancing the efficiency of operating a marine gas turbine plant (MGTP) without compromising the reliability of the plant's components is a perpetual undertaking for design bureaus and enterprises engaged in the manufacture, repair, and modernisation of gas turbine engines (GTE) incorporated within MGTP. The elements of GTE which are subject to the greatest loads (both dynamic and thermal) are those in which the combustion of organic fuels is carried out and which are under the influence of hot gas flow. Primarily, these elements pertain to the combustion chamber (CC) and the turbocharger.

The enhancement of the efficiency of contemporary GTE is accomplished through a reduction in the temperature of the intake air at the inlet to the GTE, an improvement in the quality of fuel combustion, an increase in the temperature of the gas stream at the inlet to the turbine, and an optimisation of the processes that underpin the functioning of engine elements.

In the studies [1-3], considerable attention was paid to the issue of additional cooling of the air temperature at the inlet to the GTE. A range of system and installation schemes for moderate and deep air cooling using exhaust gas products was put forward. Research findings indicate that, contingent on the climatic conditions prevalent at an MGTP, the implementation of compact coolers at an elevated level of productivity is a viable proposition. Moreover, a reduction in the air inlet temperature to 283 K (and, in certain instances, to 280 K) has been demonstrated to lead to substantial fuel savings, concomitant with an augmentation in the efficiency of gas turbine units by 10-12% [1]. Furthermore, these findings are pertinent to the context of analysing the reliability of data on the efficiency of GTEs obtained by modelling and actual research observations.

One of the ways to increase the efficiency of GTE is to increase the temperature of the gas stream at the inlet to the first stage of the high-pressure turbine [4]. It has been demonstrated that an augmentation in the temperature of the combustion products flow by 373 K can result in an 8-13% increase in power output and a 2-4% increase in the efficiency of the simple cycle. The augmentation of temperature can be facilitated by the intensification of combustion processes within the CC or, alternatively, by employing an exhaust gas recirculation system. It is important to note that the temperature of the hot gases emanating from the combustion chamber of contemporary gas turbines frequently exceeds the melting point of the alloys utilised in the nozzle and rotor blades. Constant exposure to such extremely high-temperature environments can lead to thermal fatigue of the blades and turbine failure [5-6]. Consequently, it is imperative to implement an effective cooling system for high-pressure turbine (HPT) components, in conjunction with methodologies aimed at enhancing the aerothermal characteristics of the CC-HPT assembly.

The utilisation of advanced convection-film cooling systems, in conjunction with contemporary heat-resistant materials and thermal barrier coatings (TBC), ensures that an acceptable temperature level is maintained. However, even in these instances, the cooling air consumption can account for 20-30% of the airflow through the compressor. This phenomenon impacts the thermodynamic processes within the compressor, consequently diminishing the efficiency of the GTE in its entirety. Consequently, there is ongoing work to modernise cooling systems by employing novel approaches or enhancing existing ones. For instance, effusion and transpiration cooling systems have been proposed [5], which enable cooling of the CC surface due to complete coverage of the entire hot surface with cooling air. In addition, approaches to find a balance between cooling efficiency and the aerodynamics of the external and internal contours of the blade apparatus, whilst minimising the use of cooling air, are described in [7-8].

In [9], a study was conducted to investigate the effect of the intensity and direction of cooling air swirling on the temperature state of fuel assembly elements. A range of options for the direction and angle of air swirling were put forward and mathematical models were constructed. However, experimental evaluations of this phenomenon are scarce, particularly in terms of determining the cooling efficiency of the fuel element. The advent of more stringent environmental protection requirements has led to the widespread adoption of lean mixture combustion with premixing in the design of combustion systems for GTE, a practice that exerts a significant influence on the distribution of hot flow, swirls, and turbulence at the CC outlet. In turn, this has ramifications for the design of the cooling system of the HPT stage.

The Computational Fluid Dynamics (CFD) method is currently being actively used to model processes in low-emission combustion chambers [10]; various approaches are employed to intensify the processes in such combustors, including new methods for enhancing fuel–air mixing and combustion [11, 12], particularly through the use of low-temperature plasma [13, 14].

Alongside the utilisation of software for modelling, the most significant phase in evaluating the efficacy of the cooling system for turbine components is the process of conducting actual measurements of temperatures, pressures, flow rates, and other pertinent parameters. To facilitate this research, a range of measurement systems is employed. The selection of measurement method is contingent upon the conditions under which the research objects are situated, with the distinction being made between contact and non-contact, and automatic and semi-automatic techniques [15-17]. The selection of measurement method is pivotal in determining the subsequent selection of primary transducers.

The present article is devoted to determining the thermal condition of MGTP turbine elements. The process of temperature measurement is, therefore, a focus of particular attention. Non-contact temperature measurements are typically performed using optical pyrometers [15] and thermal imagers [16], while contact measurements are accomplished through the use of thermoelectric transducers (thermocouples)

[17]. The utilisation of these primary transducers is not without its advantages and disadvantages, concurrently establishing the foundations for the configuration of the measurement system in its entirety [15-17]. The measurement of turbine element temperatures in engines is a complex process that requires the consideration of various operating conditions. Of particular complexity is the measurement of temperatures of rotating elements when using thermocouples. To address this challenge, telemetric measurement systems have been employed, which integrate primary transducers with modules for the processing of measured values. Several works [18-19] have described various approaches in the realisation of such systems, indicating their advantages and disadvantages, as well as peculiarities of their application in industry. However, there is a paucity of literature describing the operation of a telemetric measurement system (TMS) for the contact measurements of temperatures of rotating elements of the turbine.

The proposed experimental method for determining the thermal condition of MGTP elements constitutes an innovative approach to monitoring the thermal condition of engine elements during operation, as well as analysing the correctness of previously developed mathematical models and preliminary thermal calculations. The proposed TMS facilitates the acquisition of actual temperature values in engine components, thereby substantiating or refuting the precision of the modelling. Additionally, it aids in the identification of factors that have been overlooked in the calculations and exerts influence on the temperature distributions during engine operation.

PROBLEM STATEMENT

The continuous enhancement of the efficiency of marine stationary turbine plants is challenged by the necessity to maintain the reliability and longevity of critical gas turbine engine components, particularly those subjected to extreme thermal and mechanical loads, such as the combustion chamber and the high-pressure turbine. Despite advancements in cooling technologies and materials, existing methods (including convection-film cooling and the use of thermal barrier coatings) still require a significant portion of compressor air, thereby reducing overall engine efficiency. Moreover, while numerical modelling and simulation provide insights into temperature distributions, there remains a lack of comprehensive experimental data, especially concerning the thermal behaviour of rotating turbine elements under operational conditions. The current scarcity of validated telemetric measurement systems for contact temperature sensing in rotating components further limits the ability to accurately assess and optimise cooling strategies. This research seeks to address this gap by developing and implementing an experimental telemetric measurement system aimed at accurately determining the thermal state of GTE turbine components, thereby enabling the validation of thermal models and optimisation of cooling performance without compromising system efficiency or reliability.

COOLING SYSTEM PERFORMANCE ANALYSIS

It is acknowledged that the pre-twisting of cooling flow is facilitated by the pressure drop induced by the air pressure in the cavity situated in front of the HPT disc, which is analogous to the gas pressure experienced behind the nozzle blade in the root section. For a 25 MW GTE, the pressure drop can reach approximately 1.0 MPa when the ratio of inlet pressure P_{in} to outlet pressure P_{out} is between 1.9 and 2.0.

The decrease in air temperature is defined as the difference between total temperatures in absolute and relative motion, as calculated by Eq. (1) [20]:

$$\Delta T = T^* - T_w^* = \left(T + \frac{C^2}{2c_p}\right) - \left(T + \frac{W^2}{2c_p}\right) = (C^2 - W^2)/2c_p \quad (1)$$

where T is the static temperature; T^* and T^*_w are the total temperatures in absolute and relative motion; C_p is the specific heat capacity of air; C is the velocity of gas in absolute motion; and W is the velocity of gas in relative motion.

However, it is often recommended that only the circumferential components of velocities are considered in calculations, for two reasons. Firstly, the increased size of the near-disc air inlet cavity results in a greater need for consideration. Secondly, the change in the direction of flow in this cavity necessitates consideration of the effect on velocity. The resulting equation is:

$$\Delta T = \frac{(C_u^2 - W_u^2)}{2\,c_p} = \frac{(C_u^2 - (U - C_u)^2)}{2\,c_p} = \frac{2U\,Cu - U^2}{2\,c_p} \quad (2)$$

where C_u is the circumferential component of velocity in absolute motion; W_u is the circumferential component of velocity in relative motion; and U is the circumferential velocity of the RB.

Therefore, the decrease in air temperature in relative motion is found to be directly proportional to the circumferential component of velocity; that is to say, it increases continuously with increasing twist (i.e. the circumferential component of velocity). In the absence of pre-twisting, the air temperature at the disc inlet in relative motion concurrently exceeds the total air temperature at the extraction point by the value $\frac{U^2}{2\,C_p}$. In this scenario, the reduction in air temperature (ΔT) assumes a negative value at $\frac{C_u}{U} = 0.5$ (in the absence of air heating), and only in the case of $\frac{C_u}{U} > 0.5$ does the temperature reduction become evident.

To analyse the influence of other factors on the value of temperature decrease, it is necessary to take into account the air velocity at the outlet of the twisting grid inside the rotor blades of the high-pressure turbine, in the form of dependence:

$$C = \sqrt{\frac{2(P_{in} - P_{out})}{\xi \rho_{avr}}} \quad (3)$$

where ξ is the hydraulic resistance coefficient of the TG. In the case of a compressible medium (air), the resistance coefficient should be brought to the average flow density ρ_{avr}, defined as $\rho_{avr} = \frac{P_{in}+P_{out}}{2RT}$.

It is imperative to consider the available pressure drop and its hydraulic resistance coefficient when determining the velocity of airflow from the TG. Even in the presence of minimised hydraulic losses (i.e. the absence of inlet and friction losses, as well as uniform jet velocity at the outlet), the maximum theoretical flow velocity is as follows:

$$C_{max} = \sqrt{2\frac{k}{k-1}RT\left[1 - \left(\frac{P_{out}}{P_{in}}\right)^{\frac{k-1}{k}}\right]} \quad (4)$$

where k is the isentropic index and R is the universal gas constant.

Subsequently, the minimum coefficient of hydraulic resistance, which is determined by the average density, is:

$$\xi_{min} = \frac{\left(1 - \frac{P_{out}}{P_{in}}\right)}{\frac{k}{k-1}\left(1 - \left(\frac{P_{out}}{P_{in}}\right)^{\frac{k-1}{k}}\right)\left(1 + \frac{P_{out}}{P_{in}}\right)} \quad (5)$$

For the available differential pressure $\frac{P_{in}}{P_{out}} = 1.95$, the following values are obtained: C_{max} = 507 m/s and ξ_{min} = 1.05. Consequently, the maximum reduction of air temperature in relative motion is ΔT_{max} = 115 K. The actual values of flow swirl and temperature reduction are less than their maximum values due to:

- the presence of hydraulic losses at the inlet, friction losses, and the non-uniformity of velocity at the outlet ξ > ξmin;
- the presence of an axial component of velocity at the outlet of the grid $C_u = C \cdot c\, os\alpha$, where α is the angle of flow exit from the grid;
- reduction of the flow twist in the inlet cavity due to the presence of a stationary wall, lower circumferential velocity of the disc, and the difference in the diameters of the location of the twist grating and the inlet to the holes in the HPT disc. The latter causes the circumferential component of the flow velocity at the inlet to the disc hole to be less than the circumferential velocity of the flow at the outlet from the twist grating ($\frac{C_{U_{hole\ in}}}{C_u} < 1$). The phenomenon of air flow with a different degree of twist through the labyrinth seal is observed, which limits the air inlet cavity;
- air flow with a different degree of swirl through the labyrinth seal limiting the air inlet cavity.

RESEARCH OBJECTIVE

The aim of this experimental study is to demonstrate the practical application of the developed telemetry measurement system for evaluating the thermal condition of turbine components as part of the modernisation of an MGTP, with the ultimate goal of improving efficiency by raising the gas temperature at the inlet to the high-pressure turbine.

EXPERIMENTAL SETUP

The test stand employed for the research work comprised a low-pressure compressor driven by a freestanding gas turbine engine, an HPC-HPT gas generator (with cooling air supply through the TG inside the RB of the HPT), a CC, and a low-pressure turbine. The rotor and stator components of the TMS were installed in the area of the low-pressure compressor–high-pressure compressor adapter, with thermocouple wires from the RB and the HPT disc leading to the rotor part of the TMS.

To perform the thermometry, a system for recording the readings of the rotating thermocouples of the TMS was developed, based on a non-contact method of signal transmission from the rotor part to the stator part, allowing simultaneous temperature measurement by 180 thermocouples.

The rotor blades of the high-pressure turbine were sectioned according to the developed scheme. Fragments of the preparation scheme are shown in Figs. 1 and 2.

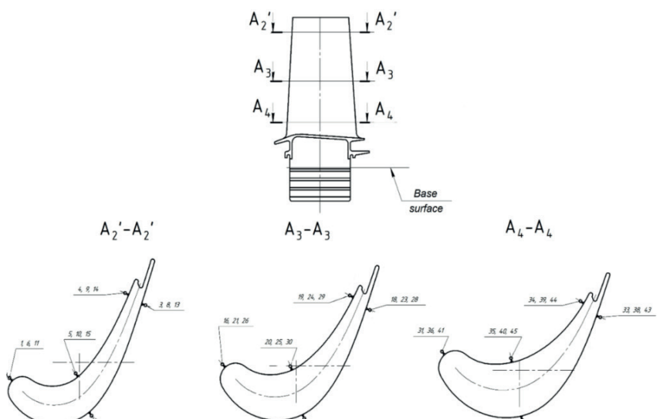

Fig. 1. The layout for thermocouple instrumentation of the rotor blade profile on the gas side

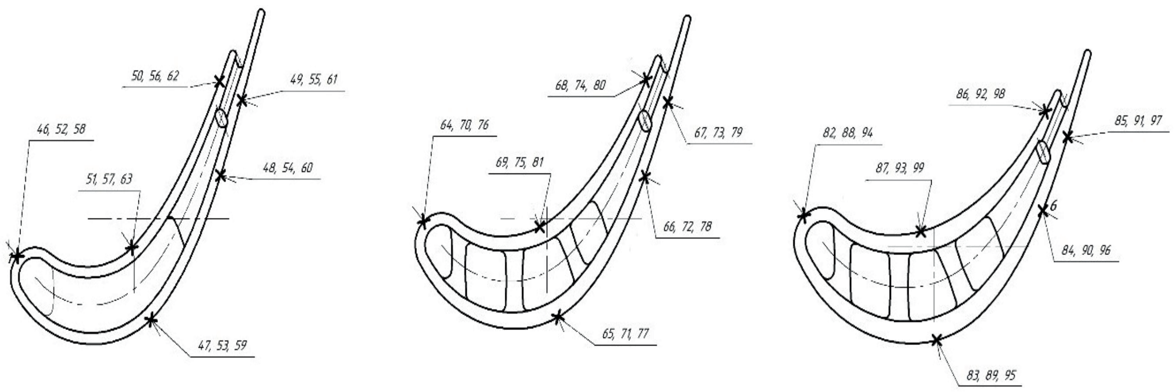

Fig. 2. The layout for thermocouple instrumentation of the rotor blade profile on the metal side

The preparation of the rotor blades on the gas side was carried out by placing thermocouples along the blade profile, followed by foil covering. The thermocouple leads were routed to the front face of the HPT disc. The preparation of the rotor blades on the metal side was carried out using micro-cable thermocouples (MCTCs) with a diameter of 0.5 mm, placed in pre-made grooves. The output of the MCTCs was conducted through the holes in the flanges, into the cavities of the blade legs and further to the back side of the HPT disc.

The preparation of near-disc cavities with rotating and fixed thermocouples, as well as pressure measurement tubes, was performed according to the scheme, a fragment of which is shown in Fig. 3.

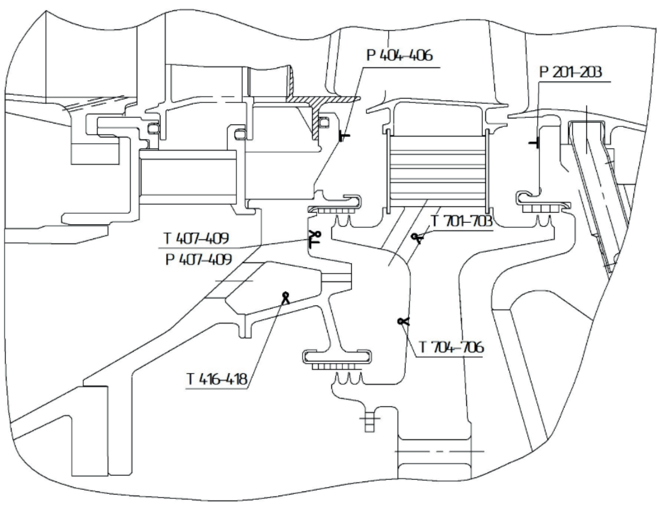

Fig. 3. Scheme of preparation of the circumferential disc cavities of the HPT

Table 1 shows the numbers of the machined rotor blades (according to the slot in the disc) and the numbers of the thermocouples installed on them.

Tab. 1. Schematic representation of recommended installation locations for rotating thermocouples

Parameter	Section	RB number in the HPT disc	Thermocouple number
Gas temperature	Peripheral	41	1, 2, 3, 4, 5
		46	6, 7, 8, 9, 10
		85	11, 12, 13, 14, 15
	Middle	3	16, 17, 18, 19, 20
		40	21, 22, 23, 24, 25
		84	26, 27, 28, 29, 30
	Root	4	31, 32, 33, 34, 35
		39	36, 37, 38, 39, 40
		83	41, 42, 43, 44, 45
Metal temperature	Peripheral	5	46, 47, 48, 49, 50, 51
		47	52, 53, 54, 55, 56, 57
		48	58, 59, 60, 61, 62, 63
	Middle	6	64, 65, 66, 67, 68, 69
		7	70, 71, 72, 73, 74, 75
		49	76, 77, 78, 79, 80, 81
	Root	8	82, 83, 84, 85, 86, 87
		50	88, 89, 90, 91, 92, 93
		51	94, 95, 96, 97, 98, 99

MEASUREMENT SYSTEM

The telemetry measurement system was designed for the primary processing and transmission of measurement signals from thermocouples between rotating and stationary bodies during thermal condition studies in GTE units and their parts. The structural scheme of the applied TMS is presented in Fig. 4.

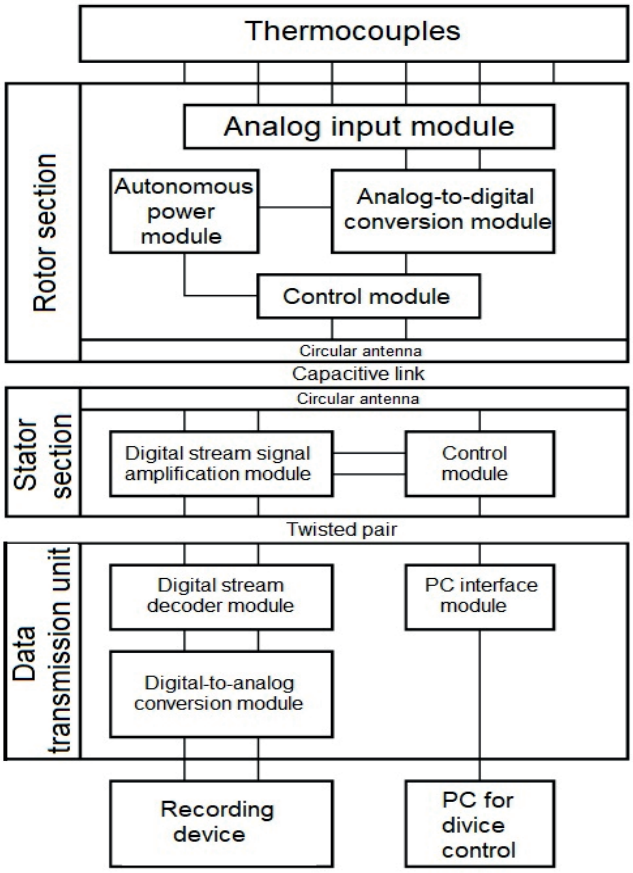

Fig. 4. Structural diagram of the TMS

The transmission of signals was facilitated by capacitive coupling, with thermocouples connected to the input of the analogue input modules. The outputs of these modules were then connected to analogue-to-digital conversion modules, which subsequently formed a digital stream for transmission to the transmitter ring antenna. The receiver ring antenna was connected to a digital stream amplification module and the amplified digital data stream was then fed to a normaliser and a differential converter. The output from the stator part of the device was fed through a twisted pair cable to the input of the digital stream decoder module of the data transmission unit. The decoded digital signal was transferred to the branched digital-to-analogue conversion modules and then the analogue measurement signal was fed to the input of the recording equipment. The rotor component of the system was powered by a battery, while the processors of the rotor and stator parts were controlled via a radio channel.

The transmission of measurement signals was conducted in real-time mode. The transmission and reception components of the antenna (capacitor linings) were isolated from the housing, which was constructed in the form of rings and installed in the rotor and stator parts of the GTE units. The power batteries, analogue input, analogue-digital conversion, and control modules were installed symmetrically in special cavities around the circumference of the rotor part housing. The modules responsible for digital stream signal amplification and control were installed asymmetrically in special cavities of the stator part's housing.

The data transmission unit, a freestanding device located outside the motor, can be positioned at a distance of up to 50 m from the motor. Control of the unit and the transmission of measurement signals was carried out sequentially.

EXPERIMENTAL INVESTIGATION

To compare the measured temperatures of the HPT rotor blades with previously calculated values, and to assess the effectiveness of the rotor blade cooling system, the measured temperatures were adjusted to the design parameters of the engine's nominal operating mode with a power output of 25 MW. Adjustment of the measured gas and rotor blade temperatures to the engine's nominal operating mode was carried out by interpolating graphical dependencies to the regime in which the gas temperature behind the high-pressure turbine nozzle guide vanes is $T = 1181 + 303 = 1484$ K (where 1181 K is the design value of the working fluid temperature behind the HPT nozzle guide vanes).

The acquisition of all data presented in this study was facilitated by the implementation of a TMS, which enabled the concurrent measurement of temperature across the fixed engine components (specifically cavities) and the rotating components, including discs and blades. The TMS under discussion makes it possible to monitor the thermal condition of these parts simultaneously with a single device and with high measurement accuracy. The accuracy of temperature measurement when using TMS is contingent on the type of primary transducer (thermocouple) employed. In the aforementioned research, thermocouples of the chromel-alumel type were utilised, resulting in a temperature measurement error of less than 1%. Such a measurement error value was achieved due to the use of high-precision temperature measurement modules with an instrumental error not exceeding 0.2% within a wide temperature range up to 1600 K, as well as through the application of the chromel–alumel thermocouples of class 2 with a measurement error not exceeding 0.75%.

RESULTS

TEMPERATURES OF GAS FLOWING THROUGH THE ROTOR BLADES

The results of the gas temperature measurements flowing through the HPT rotor blades are shown in Fig. 5 and they indicate the temperature distribution along the blade profile, which is significantly non-uniform. As shown in Table 1, for each cross-section, three blades were instrumented with thermocouples, with identical positioning of the temperature measurement points. In Fig. 5, the arithmetic mean values of temperatures from three different RB are presented at each measurement location. It should be noted that, due to the failure of certain thermocouples (caused by the aggressive environment), the mean value was, in some cases, calculated based on two rather than three thermocouple readings. The reported temperature values are reduced to the design parameters of the nominal operating mode of a 25 MW engine, which were confirmed during repeated tests (thermal performance evaluations) of the engines within their serial production process.

The maximum temperatures are observed on the concave side, with values of 1528, 1592, and 1477 K in the peripheral, middle, and root sections, respectively. Conversely, the minimum temperatures are obtained from the convex side of the profiles and are 1156, 1394, and 1095 K, respectively. Concurrently, the radial averaged gas temperatures in these cross-sections are 1381, 1396, and 1330 K, respectively. During preliminary calculations of the temperature state of the RB, these values were maintained as constants for the corresponding cross-section.

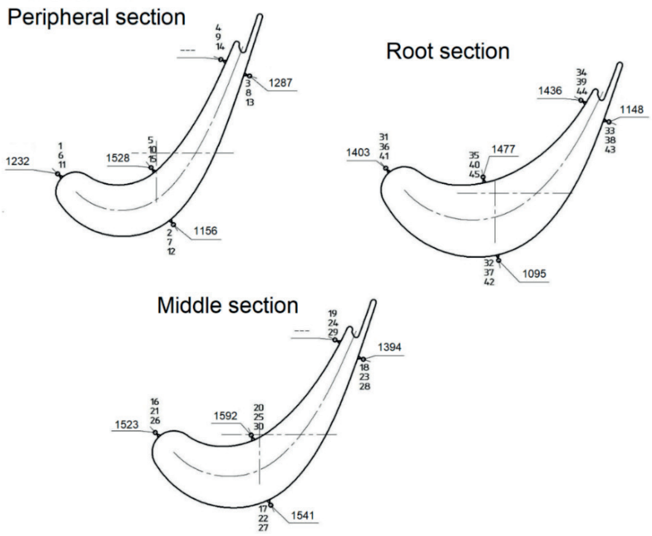

Fig. 5. The gas temperatures flowing through the HPT rotor blades have been adjusted to the rated power operating mode

The distribution of gas temperatures can be attributed to the temperature segregation of the flow for the RB, resulting from the non-uniformity of the gas temperature behind the combustion chamber and the incomplete mixing of the cooling air. The cooling air recirculates within the flowing part, in the area of the NGV, through the annular gaps between the stator and rotor in front of the RB of the HPT, as well as the exhaust of the air intended to cool the NGV. The indicated maximum gas temperatures slightly exceed the gas temperatures at the combustion chamber outlet without their reduction, due to mixing with the cooling air flowing out of the NGV of the HPT and taking into account the recalculation of parameter changes in relative motion. Consequently, the validity of the measured gas-side temperatures requires further experimental substantiation.

The gas temperatures, averaged over five points in the peripheral section of the blades, are below the design value, the temperature averaged over the middle section exceeds the design value, and the temperature averaged over the root section is, again, below the design value. This may be a consequence of the increased non-uniformity of the gas temperature distribution compared to the design value.

METAL TEMPERATURE MEASUREMENTS

The results of the RB metal temperature measurements are presented in Fig. 6. The presented temperatures were calculated in the same manner as the above-described gas temperatures. It is evident that, at most points along the profile, the measurements were obtained using a single thermocouple. In instances where two or three thermocouples were employed, the observed difference in readings ranged from 2 to 95 K, with the majority falling within a 25 K range. The maximum temperatures of the RB metal are recorded at the entrance edges of the middle section, with temperatures reaching 1216 K. In the root section, the metal temperatures are found to be comparable to those at the entrance edge, owing to the high gas temperatures on the concave side. However, in the peripheral section, these temperatures reach 1220 K, which exceeds the values recorded at the entrance edge.

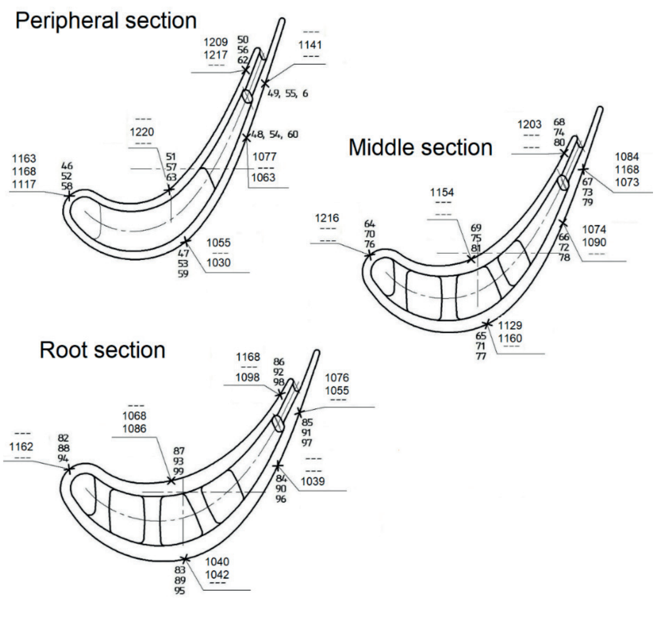

Fig. 6. The metal temperatures of the rotor blades adjusted to the rated power operating mode

DISCUSSION

In order to verify the reduction of cooling air temperature at the inlet to the RB due to pre-twisting, a preliminary modelling of the aerodynamics of the air supply path to the RB of the HPT was performed using the ANSYS CFX computational complex. The geometrical model incorporated the channels of the sub-twisting grating (comprising a segment of the antecedent cavity), the air inlet cavity constrained by labyrinth seals, and an aperture in the disc for the provision of cooling air to the RB. The modelling performed confirmed the decrease in the total air temperature in relative motion, compared to the total air temperature at the inlet of the sub-twisting grille. The results of the simulations indicated an air temperature reduction of $\Delta T = 75$ K, while the maximum theoretical value of the temperature drop, determined by the maximum theoretical flow rate and without considering swirl losses in the inlet cavity, was $\Delta T_{max} = 115$ K.

The experimental studies conducted on the difference of readings of the corresponding thermocouples ($T_{701-703} - T_{416-418}$) revealed that the decrease in air temperature in relative motion was not recorded (see Fig. 3). Consequently, the temperature measured with rotating thermocouples ($T_{701-703}$) in the holes of the disc is, on average, 2–5 K higher than the air temperature in front of the NGV ($T_{416-418}$) and the difference between these temperatures remains constant. Concurrently, the air temperature in the upper section of the inlet cavity, situated behind the NGV of the HPC ($T_{407-703}$), is approximately 17 K higher than the temperature in the front section of the NGV of the HPT. This discrepancy is probably attributable to the leakage of hot air from the cavity situated in front of the locking joints of the HPT disc. It is important to note that this temperature does not accurately reflect the mean air temperature in the cavity situated behind the NGV of the HPT. It was not possible to obtain stable air temperature measurements in relative motion in the lower part of the inlet cavity ($T_{704-706}$) during the tests.

The application of the proposed TMS facilitated the evaluation of the effectiveness of the measures developed on the basis of the modelling carried out, with the objective of reducing the air temperature of the cooling system of the first-stage HPT cooling system. It is noteworthy that the RB metal temperatures measured in the conducted studies exceed both the calculated values and the results of separately conducted thermal-hydraulic testing of blades on specialised stands for assessing the cooling system efficiency. This discrepancy is attributed to a substantial excess in the measured gas temperature relative to the design temperature. The reason for the absence of gas temperature reduction in relative motion during these tests has not been definitively established. It is likely related to the increased annular gap above the channels of the swirl grid and, consequently, to a reduced swirl intensity in the supply cavity. In addition, the lack of temperature reduction in relative motion may also be caused by increased leakage of hot air into the supply cavity. To further investigate this issue, it is recommended that the methodology for calculating the temperature state of the RB of high-temperature GTEs (with large cooling air returns) be revised. This should include taking into account the segregation of gas temperature along the profile by performing CFD modeling of the stator–rotor segment, considering the non-uniformity of the gas temperature downstream of the combustion chamber and the inflowing cooling air.

CONCLUSIONS

The experimental measurement of metal and gas temperatures in contact with the blades and high-pressure turbine disc of a gas turbine engine was carried out using the proposed innovative telemetry measurement system. The obtained temperature data allowed for a detailed assessment of the thermal state of the turbine components and the extent of temperature reduction in the cooling air in relative motion, due to swirling in the nozzle vanes of the HPT. The developed TMS demonstrated high accuracy in measuring both the working fluid and blade metal temperatures, with a margin of error of less than 1%. This precision enabled the identification of factors influencing the thermal distribution within engine components that were previously unaccounted for in mathematical simulations. The introduction of the proposed TMS is expected to support an increase in gas temperature at the turbine inlet, ultimately contributing to improved efficiency of marine gas turbine plants.

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