UDC 621
DOI: 10.15587/1729-4061.2025.337285

DEVISING A METHOD FOR DESIGNING MULTICOMPONENT DIFFUSERS OF COMPRESSORS IN TURBOGENERATORS WITH HYDROGEN COOLING

This study's object is the aerodynamic characteristics of the compressor diffusers in hydrogen-cooled turbogenerators.

This paper reports a solution to the task of improving the efficiency of a cooling system discharge unit. This issue relates to enabling the necessary gas consumption while reducing the number of diffuser blades. The problem was solved by introducing a seal with improved technological control capabilities and step-by-step optimization of the diffuser flow part.

Another issue is the thermal loads on the power components of the diffuser unit due to the action of circulating currents and temperature gradients. The problem was solved by introducing dielectric and non-magnetic elements into the structure, by additional finning and multi-component design for the diffuser strength circuits.

The main result is the designed multi-component diffuser structure with a number of blades of 20 while enabling a head of H_{st} = 978 mm H_2O. The adopted duct opening angle was 20°. Other necessary geometric parameters were determined. The introduction of a multilayer seal made it possible to reduce the gap between the impeller and the diffuser to 0.9 mm. The proposed design was tested on the bench at a manufacturing enterprise.

The results of the study are attributed to the use of non-magnetic and dielectric materials (AISI 321 steel and fiberglass), as well as the introduction of additional strength decoupling elements.

A special feature of the proposed method is the application of mathematical models based on the basic equations of gas dynamics, taking into account the composition principles of engineering alloys and synthetic materials. This could be achieved via a step-by-step optimization of the design.

The proposed structure could be implemented when designing and modernizing hydrogen-cooled turbogenerators

Keywords: turbogenerator, compressor diffuser, multilayer seal, gas-dynamic calculations, circulating currents

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Received 07.06.2025
Received in revised form 28.07.2025
Accepted 11.08.2025
Published 26.08.2025

How to Cite: *Tretiak, O., Kravchenko, S., Shestak, B., Shpitalnyi, D., Arefieva, M., Tretiak, I., Serhiienko, S., Kovryga, A. (2025). Devising a method for designing multicomponent diffusers of compressors in turbogenerators with hydrogen cooling. Eastern-European Journal of Enterprise Technologies, 4 (8 (136)), 26–38. https://doi.org/10.15587/1729-4061.2025.337285*

1. Introduction

Enabling trouble-free operation of turbogenerators remains a problem that has not yet been fully resolved throughout the entire period of operation of electric machines. The reliability of a turbogenerator is significantly affected by the trouble-free operation of its cooling system [1–3]. Insufficient efficiency and reliability of the cooling system can cause turbogenerator failures, which almost always leads to long downtimes and high repair costs [4, 5].

When designing any electric machine, it is necessary to comply with strict requirements stipulated by regulatory documents regulating the thermal state of active and structural parts. The decisive influence in this is the choice of the cooling system and, above all, the type of pressure or exhaust elements (axial or centrifugal fans). Enabling a given cooling gas flow rate Q for a given geometry of the cooling channels, i.e., with constant hydraulic head $Z=\text{const}$, requires the corresponding fan pressure $H=Z\cdot Q^2$. One way to increase pressure H is to convert part of the dynamic pressure after the impeller into static pressure. This transformation can be obtained in a diffuser located behind the impeller. With appropriate geometric parameters characterizing its flow part, the diffuser also significantly contributes to increasing the aerodynamic efficiency of the compressor as a whole.

One of the most important geometric parameters of the compressor is the diameter, shape, and number of blades in the blade diffuser, as well as their installation angles. It should be noted that a violation of any of the geometric parameters, leading to a deviation from its close to optimal value, could lead to serious consequences. Namely, to a violation of the flow structure, the occurrence of reverse flows in the channel, which would lead to a decrease in the aerodynamic parameters of the compressor (static pressure, efficiency, etc.). The result is a decrease in the performance of the centrifugal compressor and the associated deterioration of the thermal state of the generator.

Devising a new method for designing diffusers using mathematical models based on the basic equations of gas dynamics with step-by-step optimization will make it possible to determine the optimal parameters of their structure. Step-by-step optimization will be carried out by changing the angle of attack of the compressor wheel blades.

The main structural elements of the discharge unit are the diffuser units, shield mounts, and seals. The introduction of structures made of non-magnetic and dielectric materials into structural power components will reduce the impact of circulating currents on the magnetic elements of the turbogenerator located in the compressor area while maintaining the reliability of the structure.

Therefore, studies aimed at designing an improved structure of the compressor diffuser in high-power turbogenerators, which could simultaneously ensure reliable operation, high efficiency due to reduced heat losses, as well as ease of maintenance, are relevant.

2. Literature review and problem statement

One of the possible ways to improve the efficiency of a multi-component diffuser is to change its flow part. At the same time, for existing diffuser structures, one should strive to reduce the number of blades in order to reduce aerodynamic resistance. The introduction of an excess number of blades into the design was dictated not by aerodynamic principles but by the need to enable the required structural rigidity to reduce deformations under thermal non-uniform loads.

Works [6, 7] describe methods for designing new structures of compressors, diffusers, and their elements. It is shown that it is currently possible to switch to profile grilles with a reduced number of blades. The advantage of these papers is the detailed study of all components and the coordination of their pressure characteristics with the general system in which they operate. However, in the case of turbogenerators, it is additionally necessary to ensure the safe operation of pressure vessels. Therefore, along with the optimization of aerodynamic parameters, confirmation of guaranteed sufficient reserves for thermal load and mechanical strength is a necessary condition.

The disadvantages of those studies are the lack of experimental data for specific types of diffuser designs for high and medium-power turbogenerators. The reason for this is the lack of sufficiently large series of turbogenerators of similar capacities with the necessary structure.

The main challenges associated with improving the efficiency of the cooling system can be solved by designing new diffuser structures with a reduced number of blades with the accompanying redesign of the flow part. Currently, the dominant methods for researching structural solutions are CFD methods [8, 9]. Their disadvantage is that in order to obtain reliable results, it is necessary to set boundary conditions at sufficiently high accuracy. The reason why this is difficult to achieve is the factor that different diffuser designs are used for the same compressor structure, which leads to a spread of more than 10% for its operating points. Accordingly, all channels of the flow part must be regulated directly for each individual instance of the turbogenerator.

In [10], the influence of the profile of the subslot channel along the length of the rotor on the distribution of air speeds in the radial channels and the efficiency of heat transfer in the turbogenerator was investigated. In [11], the dependence of the change in the compressor performance for hydrogen transportation was determined. The disadvantage is that the findings cannot be fully applied due to the lack of consideration of specific parameters of the main working fluid of the turbogenerator cooling system (hydrogen with the specified purity).

In [12], an improved Taher-Evans cubic polynomial (TE-CP) method for calculating the polytropic characteristics of centrifugal compressors is presented. The emergence of significant differences between the results when using different equations of state is shown. Recommendations are given for using the same equation of state throughout the entire time scale of the project to avoid potential discrepancies. The advantage of this method is the high accuracy of calculating the polytropic efficiency of a real gas centrifugal compressor using the temperature-entropy cubic polynomial path function. However, the algorithms used cannot be implemented for a turbogenerator because of the lack of parameters for pressure losses due to the geometry of the main components.

Methods for calculating real gases for centrifugal compressors are reported in [13, 14]. They include the use of several different polytropic path equations, as well as several numerical integration methods. In [14], operational test data for centrifugal compressors are additionally given.

The disadvantage of papers [13, 14] is that no research was conducted for the case of increasing the pressure to the required value using hydrogen as a working fluid. Without additional research, these solutions cannot be extended to turbogenerators operating with an excess hydrogen pressure of 0.2...0.3 MPa. The reason is the need to use already fully assembled turbogenerators with different diffuser configurations to conduct this type of tests. For many existing turbogenerator designs, one of the current problems is the issue of circulating currents. This problem arises due to the presence of powerful magnetic fluxes created by the interaction of the electromagnetic field of the rotor and stator. It becomes especially relevant for hydrogen-cooled turbogenerators with compressor diffusers made of magnetic materials. The reason for its occurrence is eddy currents induced when an alternating magnetic field (from the rotor during its rotation) crosses the diffuser array. This phenomenon is especially pronounced if the diffuser is closed in the current circuit (stator housing – diffuser shields through bolts or welded joints – guide vanes and inner walls of the diffuser). The consequences of the action of circulating currents are local heating (especially at the edges and corners), increased thermal loads, and overheating of the insulation of the front parts near the diffuser, energy losses as a result of an increase in the temperature of hydrogen.

Another issue is the uneven thermal expansion of the elements of the diffuser structure of the turbogenerator compressor. This unevenness is caused by the action of circulating currents, blowing the diffuser with gas flows of different temperatures, the proximity of the diffuser to other heated elements (windings, stator core, etc.).

It is also necessary to consider the problem that arises when using connections of different materials in the structure of the diffuser assembly. Its cause is that different materials have different coefficients of linear expansion.

These two issues lead to uneven loading on welded joints and fastening elements due to temperature gradients. The consequences can be loss of geometry and deterioration of gas dynamics caused by deformation of the diffuser, violation of fastenings and tightness, fatigue damage during multiple starts/stops. The possibility of these damages makes it necessary to provide excess safety margins by increasing the mass and dimensions.

In work [15], the main types of turbogenerator designs are described. When designing new elements for these structures and upgrading existing ones, options for replacing the materials of the elements with non-magnetic and synthetic ones are presented. In [16], a program for calculating the performance of a centrifugal compressor is given, based on the method of running consumption over time.

The disadvantage of studies [15, 16] is the lack of a principle of criterion-based improvement of individual components due to the small standard sizes of the components, for which thermal deformations are not the dominant factor of the force solution. The reason is that modern research on the introduction of new materials into the design is mainly carried out in the field of alternative energy, the generating capacities of which are relatively small (compared to turbogenerators). However, the application of these technological advancements in classical energy requires adaptation of the proposed solutions. As an example of the design features of high-power turbogenerators, significant temperature stresses and uneven loads on welded joints and fastening elements due to temperature gradients can be noted.

Summarizing the above, as well as taking into account the content of works [7, 11, 13], which elaborate in detail the methods for modernizing injection systems, the following conclusion can be drawn.

High-power turbogenerators may switch to more efficient cooling systems by modernizing the structure of guide devices and multi-component diffusers. But the problem is the lack of methods for determining the optimal geometry of diffusers, the design of which uses modern synthetic materials, while enabling the specified aerodynamic parameters for high-power turbogenerators with hydrogen cooling. This is due to the fact that there is currently no regulatory framework for the use of these materials for such units. There is also no algorithm for calculating multi-component diffusers that combines the use of theoretical and experimental data and effectively takes into account the complex mutual influence of the set of parameters on which the aerodynamic characteristics depend, with their step-by-step adjustment.

3. The aim and objectives of the study

The aim of our research is to devise a new method for designing diffusers using mathematical models based on the basic equations of gas dynamics, taking into account the principles of composition of engineering alloys and synthetic materials, with step-by-step optimization of the structure. This will make it possible to improve the efficiency of the design of the discharge unit by reducing aerodynamic resistance along the flow part, the mass of the unit itself, and the effect of thermal factors due to the reduction of circulating currents.

To achieve the goal, the following tasks were set:

– to conduct a study on the dependence of efficiency of the compressor diffuser structure on the geometric parameters of the flow part and the number of blades;

– to design a structure of a multi-component diffuser with a minimum number of blades, which enables the required gas flow;

– to design the structure and manufacturing technology for the compressor diffuser power belt, which makes it possible to reduce the phenomenon of non-uniform deformations caused by thermal expansions and significantly reduce the phenomenon of circulating currents.

4. The study materials and methods

The object of our study is the aerodynamic characteristics of diffusers in high-power hydrogen-cooled turbogenerator compressors. The principal scientific hypothesis assumes that the diffuser circuit is a closed system. The head characteristic of the fan is used as the initial parameter. The output parameters are the aerodynamic resistances of the turbogenerator cooling system. It is required to provide the required vector and level of cooling gas speeds by profiling the working part of the diffuser.

The purpose of our study is to lighten the structure while maintaining its mechanical rigidity and introducing non-metallic elements to ensure a significant reduction in the influence of circulating currents.

To scientifically substantiate the effectiveness of the proposed diffuser design, a number of studies were conducted at different angles of attack and with different numbers of blades using several methods. At the same time, a study on the sensitivity of the air gap of the seal was conducted. Minimizing the gap inside the seal makes it possible to guarantee the absence of hydrogen leaks and provides normal aerodynamic characteristics.

The experimental method involved conducting measurements on operating samples of TGV series turbogenerators with a capacity of 200 MW and 325 MW.

In the experimental study, correlation analysis was used, which determined the convergence of temperature changes for the gas cooler – turbogenerator ventilation duct system – diffuser. Due to the coincidence of the temperature gradient, the convergence parameter of the general three-dimensional calculation was achieved.

The experiments were carried out on turbogenerators operating under rated load and at nominal pressure of hydrogen of a certain purity. Diffusers on different samples of turbogenerators had different numbers and angles of blade installation, as well as, accordingly, different geometry of the shape of the blades and inter-blade channels.

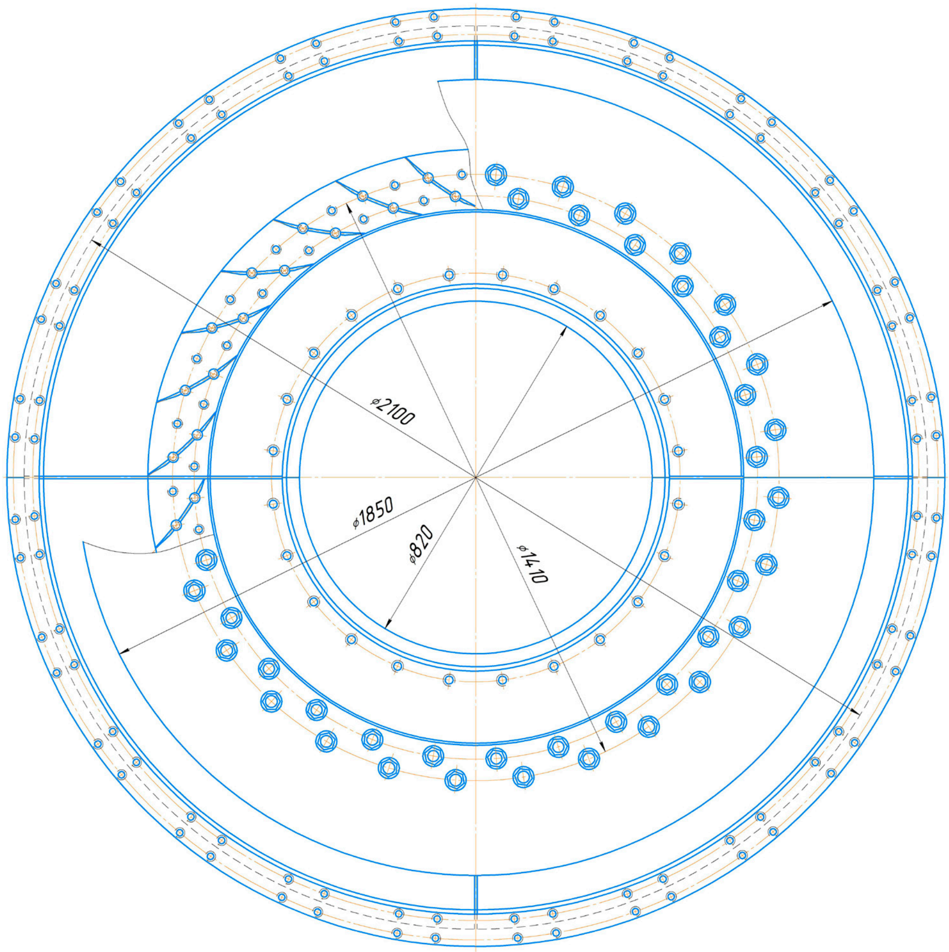

Fig. 1. Diagram of a compressor diffuser in the TGV-325 turbogenerator

The second method of our study was three-dimensional modeling using the CFD method. As a simplification, a separate calculation of the diffuser was performed: the head characteristics of the fan and the characteristics of pressure loss in the turbogenerator ventilation system were specified, which were obtained from analytical calculations.

The disadvantage of the typical diffuser design is its excessive rigidity. With a significant temperature difference, it leads to incorrect operation of the seals due to uneven rigidity of the structure. This, in turn, causes uneven deformations and, as a result, displacement of the seals. The classical solution from engineering mechanics cannot be applied in this case since a decrease in the rigidity of the main elements of the diffuser structure leads to deformations caused by the difference in hydrogen pressure inside and outside the circuit.

When designing the proposed structure, a method of composition of metal and dielectric materials was used; they have significantly different coefficients of thermal expansion. At the same time, the fastening of these elements was provided by stainless steel bolts and the additional inclusion of insulating sleeves in the structure, which make it possible to damp mechanical loads.

To confirm the operability of the designed structure, a gas-dynamic calculation was performed based on existing methods [8, 9]. The results of the calculation were correlated with full-scale tests at a manufacturing company's test benches.

5. Results of research into the proposed structure of the compressor diffuser in a turbogenerator

5. 1. Research into the efficiency of compressor diffuser design when changing its parameters

Our study aims to determine the optimal parameters for the structure of a compressor diffuser in a turbogenerator with a capacity of 325 MW manufactured by JSC "Ukrainian Energy Machines", which could make it possible to improve its efficiency. The study is based on the criteria for optimizing the flow part and choosing the number of blades.

The general view of the compressor diffuser in the turbogenerator TGV-325 is shown in Fig. 1.

When considering the issue of changing the number of diffuser blades (z_{dif})) in the TGV-325 turbogenerator, it should be borne in mind that the currently available centrifugal compressor structure has been tested by long-term operating experience. It has proven efficient at many thermal power plants under various climatic conditions.

The results of our research were obtained through a three-dimensional CFD simulation in the SolidWorks Simulation Flow software package, as well as from field studies on a compressor sample at n = 3000 rpm for diffusers:

– with a different number of blades (z_{dif} = 28…56);

– in a wide range of installation angles (α_3 = 5…24°);

– with values of the radial gap between the impeller and the diffuser (δ = 0.9…2.5 mm).

To obtain a qualitative picture of the influence of the geometric parameters of the diffuser, experimental data obtained on a full-scale model of a centrifugal compressor of the TGV-200 turbogenerator with a capacity of 200 MW were also used. The diffuser blades of this compressor are made straight, unlike the curved diffuser blades of the TGV-325 turbogenerator.

In order to determine the optimal design parameters of the compressor diffuser, an analysis was conducted on the possibility of its use with the number of blades z_{dif} = 20. Other parameters (D_3, D_4, b_3, b_4, α_3, α_4) applied were in accordance with the drawing of the compressor diffuser in the TGV-325 turbogenerator (Fig. 1). The cooling gas discharge scheme with a blade diffuser is shown in Fig. 2.

The following parameters are determined according to the scheme shown in Fig. 2:

– D_3 – diameter at the inlet to the blade diffuser (m);

– D_4 – diameter at the outlet from the blade diffuser (m);

– b_3 – width of the blade diffuser at the inlet (m);

– b_4 – width of the blade diffuser at the outlet (m);

– α_3 – angle of inclination of the blade (or flow) to the circle at the inlet to the diffuser (°);

– a_4 – angle of inclination of the blade (or flow) to the circle at the outlet from the diffuser (°);

– $R_{aver.blade}$ – radius of curvature of the centerline of the diffuser blade profile (m).

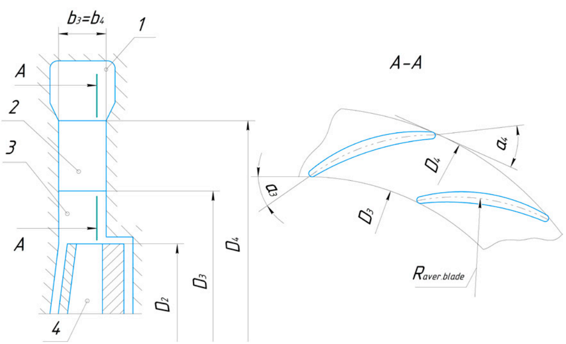

Fig. 2. Cooling gas discharge scheme with a blade diffuser: 1 – spiral assembly; 2 – blade diffuser; 3 – slot diffuser; 4 – centrifugal impeller

The radius of curvature of the centerline of the diffuser blade profile is determined from the following formula

$$R_{aver.blade}=\frac{D_4^2-D_3^2}{4\cdot\left(D_4\cos a_4-D_3\cos a_3\right)}. \qquad (1)$$

The average angle of expansion of the inter-blade channel of the diffuser (the average angle between the side walls of the channel, which determines the degree of its divergence) is determined from the following formula

$$\Theta_{aver}=\frac{180°}{z_{dif}}\cdot\left|\begin{matrix}2-\dfrac{D_3}{2\cdot R_{aver.blade}}\cdot\cos a_3-\\ -\dfrac{D_4}{2\cdot R_{aver.blade}}\cdot\cos a_4\end{matrix}\right|. \qquad (2)$$

The main task of the diffuser is to increase the static pressure of the gas due to the kinetic energy of the flow. The higher the flow velocity at the outlet of the compressor impeller ($C_{2calc}=235$ m/s), the more developed the diffuser should be. The blade diffuser makes it possible, with the same decrease in gas velocity due to an increase in radius (as in a bladeless diffuser), to additionally reduce the velocity due to forced rotation of the flow from the tangential to the radial direction. At the same time, the component of the circumferential velocity decreases and the angle of the flow exit from the compressor increases. As a result, the blade diffuser allows the gas to be compressed with smaller radial dimensions than in a bladeless device and on a shorter gas path within the diffuser.

One of the important criteria when calculating and choosing the shape of the flow part of the blade diffuser is the number of blades. To provide the required cross-section and maintain the shape of the channels formed by adjacent blades, the number of blades should be minimal.

The optimal shape of the channels formed by adjacent diffuser blades should meet the following recommended conditions:

1) for a given cross-section, the hydraulic radius of the channel should be minimal. The best practical approximation to this requirement should be a square cross-section at or near the inlet of the diffuser;

2) the diffuser channel should have straight walls of the housing;

3) the number of blades should be minimal, which satisfies the optimal shape of the channel.

The optimal length is set (best) at the lower limit of the recommended lengths. According to the basic equations from gas dynamics, it was found that the length of the inter-blade channel of the diffuser should be four times greater than the distance between the inlet edges of two adjacent blades ($l=4a$).

Diffusers with different numbers of blades are shown in Fig. 3–5.

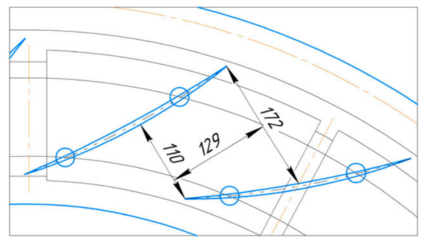

Fig. 3. Sketch of a compressor diffuser with $z_{dif}=20$

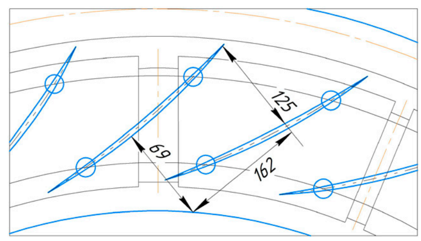

Fig. 4. Sketch of a compressor diffuser with $z_{dif}=28$

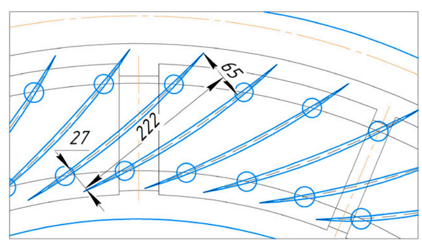

Fig. 5. Sketch of a compressor diffuser with $z_{dif}=56$

Comparing three diffusers with $z_{dif}=20$, $z_{dif}=28$ and $z_{dif}=56$ with the same blade installation angle $\alpha_3=20°$, we obtain the following data:

– for $z_{dif}=28\cdot l=4\cdot 69=276$ mm, with actual $l=162$ mm, the excess of the recommended over the actual is 1.7 times;

– for $z_{dif}=20\cdot l=4\cdot 110=440$ mm, with actual $l=129$ mm, the excess of the recommended over the actual is 3.4 times;

– for $z_{dif}=56\cdot l=4\cdot 27=108$ mm, with actual $l=222$ mm, in this case the channel length is 2 times greater than the recommended one.

The diffuser channel with $z_{dif}=56$ according to the recommendations is larger (longer) for this number of blades and installation angle. This means that with an increase in the gas flow rate in the inter-blade channel, additional losses appear, associated with the appearance of vortex zones and a change in the structure of the gas flow. For this reason, the efficiency of work for increasing the static pressure decreases, and the losses that occur when the gas flow passes through the diffuser cross-section significantly negatively affect the aerodynamic performance of the compressor. In turn, the shape (geometry) of the diffuser channel with $z_{dif}=20$ is less effective, under the same compressor operating mode, compared to $z_{dif}=28$. This is a consequence of the increase in the cross-sectional area and the decrease in the length of the inter-blade channel;

4) the expansion angle of the diffuser channels must be equal to or less than the angle set for straight diffusers with a uniform velocity at the inlet. For straight conical diffusers with large cross-sections, the expansion angle is 8°.

Let us compare three diffuser designs by the degree of expansion (diffusivity) of the interblade channel:

– for $z_{dif} = 20$: $b / a = 172/110 = 1.56$;

– for $z_{dif} = 28$: $b / a = 125/69 = 1.81$;

– for $z_{dif} = 56$: $b / a = 65/27 = 2.41$.

The recommended degree of expansion for diffusers with parallel side walls and an angle of expansion of about 8.5° is $b / a = 1.6$.

Increasing the length of the channels by more than $b / a > 4$ does not improve the operation of the diffuser since the positive effect of converting velocity into pressure is lost due to additional losses in the diffuser and vortices.

Long and curved channels cannot improve the conversion of velocity into pressure since, in this case, the flow is pushed to the wall with a smaller curvature of the channel where high velocities are restored again;

5) the selection of inlet angle α_3 is carried out after selecting the optimal geometric ratios of the dimensions of the diffuser channel.

The dependence of static pressure on the mass flow rate of the cooling gas $H_{st} = f(Q_{gas})$ is shown in Fig. 6. The diffuser parameters, at which the dependence curves depicted in Fig. 6 were constructed, are given in Table 1.

The dependence of efficiency on the mass flow rate of the cooling gas $\eta = f(Q_{gas})$ is illustrated in Fig. 7. These dependences were obtained during a full-scale study of a centrifugal compressor for turbogenerators TGV-200 and TGV-325. The diffuser parameters, at which the dependence curves shown in Fig. 7 were constructed, are given in Table 2.

The plots (Fig. 6, 7) show the results of research on the complete family of fans and diffusers with possible parameters for TGV-325 turbogenerators. A wide range of our research allowed us to evaluate the basic technical parameters, such as head and flow rate, as well as the main geometric characteristics. During further design development, the resulting family of curves has made it possible to select the parameters for the diffuser with the minimum number of blades, provided that the required gas flow rate is ensured.

The resulting curves 25 and 26 (Fig. 6) indicate that the diffuser blade installation angle $a_3 = 20°$ at a gas flow rate $Q_{gas} = 22$ m^3/s, required for cooling the TGV-325 turbogenerator, provides the maximum compressor head. This head is $H_{st} = 940 \div 960$ mm H_2O ($H_{st} = 9220 \div 9420$ Pa), and the efficiency according to curve 12 is $\eta = 0.57$ (Fig. 7).

The plot of $H_{dyn} = f(Q_{gas})$ dependence of the centrifugal compressor of the turbogenerator type TGV-325 is shown in Fig. 8. The curves on this plot are constructed based on the results of data obtained during the simulation of purges in the SolidWorks Simulation Flow software package using the CFD method. The diffuser parameters at which the curves of $H_{dyn} = f(Q_{gas})$ dependence were constructed: 1 – $a_3 = 20°$, $z_{dif} = 56$, $\delta = 2.5$ mm; 2 – $a_3 = 20°$, $z_{dif} = 56$, $\delta = 0.9$ mm; 3 – $a_3 = 22°$, $z_{dif} = 56$, $\delta = 0.9$ mm; 4 – $a_3 = 20°$, $z_{dif} = 28$, $\delta = 0.9$ mm; 5 – $a_3 = 24°$, $z_{dif} = 56$, $\delta = 0.9$ mm. The point values on the plot are obtained from the results of experimental data from tests of the centrifugal compressor at a manufacturer's bench.

According to the plot of $H_{dyn} = f(Q_{gas})$ dependence, curve 3 (Fig. 8) for $z_{dif} = 28$, $a_3 = 20°$ at a gas flow rate $Q_{gas} = 22$ m^3/s, the compressor head is $H_{dyn} = 975$ mm H_2O ($H_{dyn} = 9565$ Pa).

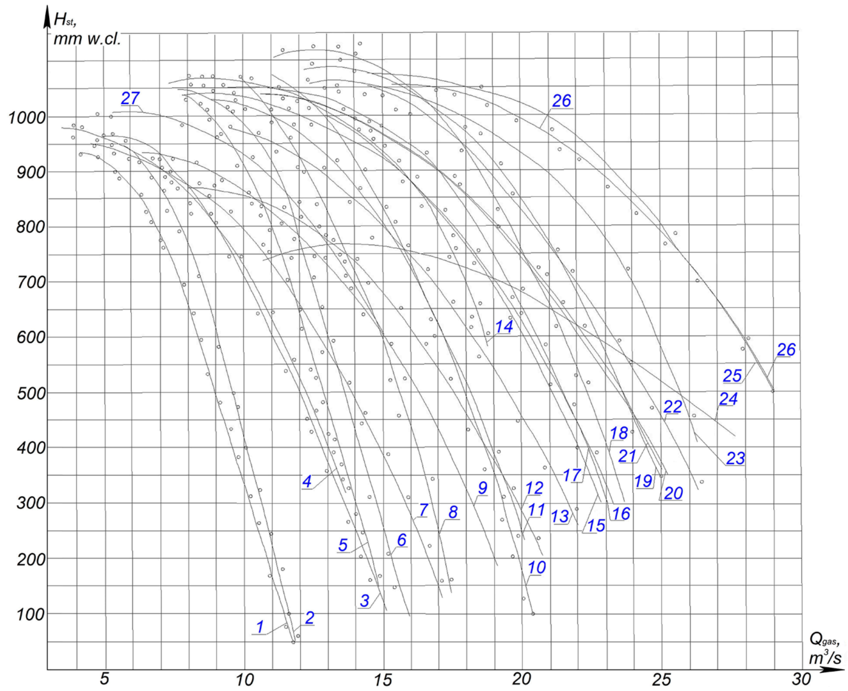

Fig. 6. Dependence plot $H_{st} = f(Q_{gas})$

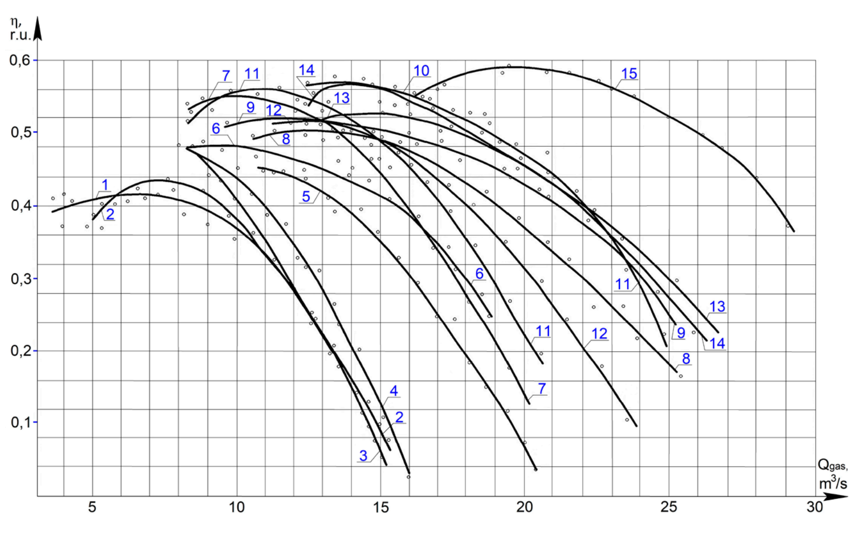

Fig. 7. Dependence plot $\eta = f(Q_{gas})$

Table 1

Diffuser parameters for which the dependence curves $H_{st} = f(Q_{gas})$ were constructed

Curve No.	Angle a_3, °	Number of blades z_{dif}, pcs.	Presence of a guiding device (blades curved against rotation)	Blade shape
1	5	56	Without the device	Curved (not shortened)
2	5	56	With the device	Curved (not shortened)
3	8	56	Without the device	Curved (not shortened)
4	8	56	Without the device	Straight
5	8	56	With the device	Curved (not shortened)
6	10	56	With the device	Curved (not shortened)
7	17	56	With the device	Straight
8	6	28	With the device	Curved (not shortened)
9	14	56	With the device	Straight
10	8	28	With the device	Curved (not shortened)
11	17	56	Without the device	Curved (not shortened)
12	16	56	With the device	Curved
13	16	56	With the device	Curved
14	12	56	With the device	Straight
15	14	56	Without a device with a narrower	Straight
16	14	56	Without the device	Curved (not shortened)
17	14	56	Without the device	Straight
18	12	28	With the device	Straight
19	16	56	Without a device with a narrower	Curved
20	16	56	Without the device	Straight
21	16	56	Without a device with a narrower	Straight
22	16	56	Without the device	Curved (not shortened)
23	12	28	Without the device	Straight
24	–	–	Without the device	–
25	20	56	Without the device	Curved (shortened)
26	20	56	Without the device	Cut by connector

Table 2

Diffuser parameters for which the $\eta = f(Q_{gas})$ dependence curves were constructed

Curve No.	Angle a_3, °	Number of blades z_{dif}, pcs.	The presence of a guiding apparatus and the direction of blade curvature	Blade shape
1	8	56	Without device (anti-rotation)	Curved (not shortened)
2	8	56	Without device (anti-rotation)	Curved (not shortened)
3	8	56	With device (against rotation)	Curved (not shortened)
4	10	56	With device (against rotation)	Straight
5	12	28	With device (against rotation)	Curved (not shortened)
6	12	56	With device (against rotation)	Curved (not shortened)
7	16	56	Without device (anti-rotation)	Straight
8	16	56	Without device (anti-rotation)	Curved (not shortened)
9	16	56	Without a device with a narrower (against rotation)	Straight
10	16	56	Without a device with a narrower (against rotation)	Curved (not shortened)
11	16	56	With device (according to rotation)	Curved (not shortened)
12	12	28	With device (against rotation)	Curved
13	12	28	Without device (anti-rotation)	Curved
14	16	56	Without device (anti-rotation)	Straight
15	20	56	Without device (anti-rotation)	Straight

Based on data on the periodic tests of the TGV-325 turbogenerator at the bench of the enterprise JSC "Ukrainian Energy Machines", the compressor head at a hydrogen purity of 97.5% was 985 mm H_2O (9660 Pa).

Therefore, according to the plot (Fig. 8), the diffuser blade installation angle $a_3 = 20°$ is the most effective when using a compressor with a gas flow rate $Q_{gas} = 22$ m^3/s. This flow rate is necessary for normal cooling of the turbogenerator. This angle is most consistent with the angle of flow exit from the compressor impeller at high gas flows and provides maximum head.

Using the three-dimensional simulation data and test results, a theoretical curve can be constructed at three points. The dependence of compressor static pressure on the number of blades $H_{st} = f(z_{dif})$ at installation angle $a_3 = 20°$ with the same blade profile geometry is shown in Fig. 9.

A detailed analysis of our study's data allows us to conclude that the compressor pressure parameters do not follow the usual linear dependence on the number of blades. Fig. 9 demonstrates that with the number of blades 28 the value of the cooling gas head reaches a local extremum. A further increase in the number of blades leads to shading of the channel. The consequence of this is an increase in the speed of the cooling gas, which leads to an increase in resistance, and the result is a drop in head.

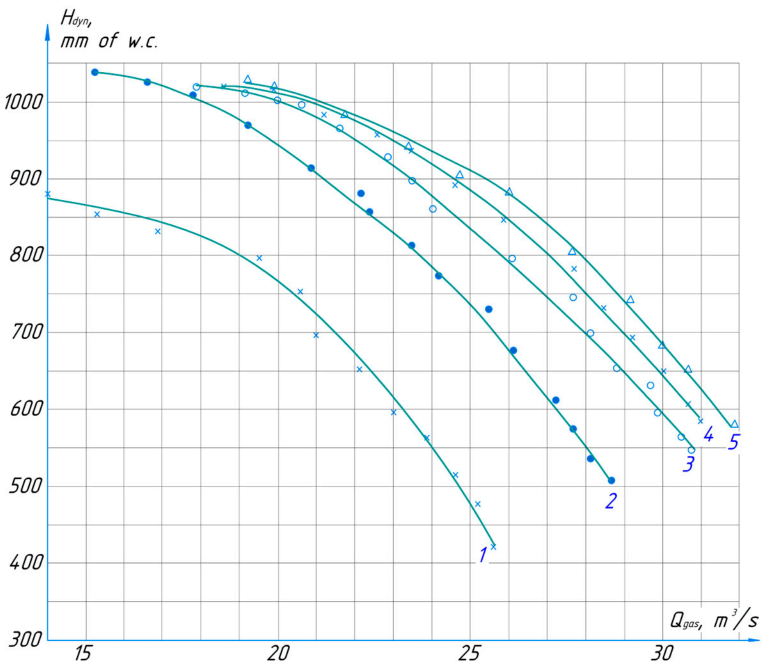

Fig. 8. Plot of $H_{dyn} = f(Q_{gas})$ dependence of the centrifugal compressor in the TGV-325 turbogenerator

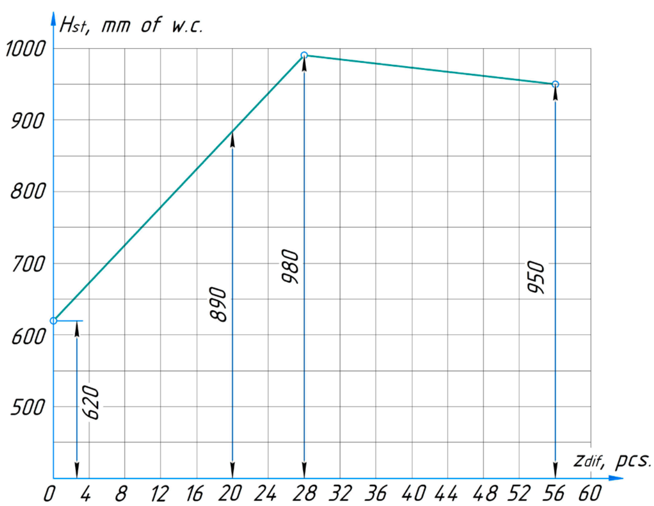

Fig. 9. Plot of compressor static head versus number of blades

Given the results above, a decision was made to use a structure with 20 blades. This number is the minimum at which it is possible to provide the necessary head characteristics of the compressor by adjusting its basic geometric parameters.

5. 2. Proposed design of a turbogenerator compressor diffuser

When designing a compressor diffuser, it is necessary to take into account that the diffuser height $\frac{(D_4 - D_3)}{2}$ or ratio $\frac{D_4}{D_2}$ (Fig. 2) are the secondary derived structural parameters. The desired shape of the diffuser channels is determined by diameter D_4 and the number of blades z_{dif}.

However, in the case of turbogenerators, based on their design and electrical features, the size of the outer diameter of the built-in compressor is strictly limited by the frontal parts of the stator winding located above it. Therefore, in accordance with the requirement of the electrical strength of the dielectric gas gap between the diffuser and the uninsulated end of the ventilation tube of the rod in the frontal parts of the stator winding of the turbogenerator, an increase in the outer diameter of the compressor (diffuser) is not possible. It must be within the maximum permissible value $D_4 = 1670$ mm, tested and approved in operation.

When designing a diffuser, it is also necessary to check the opening angle of the curved inter-blade channel in the radial plane

$$\mathrm{tg}\frac{\delta_{aver}}{2} = \frac{\pi\left(R_4 \sin a_4 - R_3 \sin a_3\right)}{z_{dif} \cdot \ell}, \tag{3}$$

where l is the length of the channel along the center line (m);

δ_{aver} is the opening angle of the equivalent straight diffuser (°).

It is recommended to choose angle δ_{aver} up to 8–10°, at least not more than 12°.

The assessment of the diffuser design according to this criterion for $z_{dif} = 20$ is based on the results of the calculation according to the following formula

$$\frac{\delta_{aver}}{2} = \mathrm{arctg}\left(\frac{\pi\left(R_4 \sin a_4 - R_3 \sin a_3\right)}{z_{dif} \cdot \ell}\right) = = \mathrm{arctg}\left(\frac{\pi\left(835 \sin 53 - 658 \sin 20\right)}{20 \cdot 121}\right) = 30°. \tag{4}$$

With the number of blades $z_{dif} = 20$, the opening angle is $\delta_{aver} = 60°$, which exceeds the recommended value by five times.

With the number of blades $z_{dif} = 28$, the opening angle is $\delta_{aver} = 34°$.

With the number of blades $z_{dif} = 56$, the opening angle is $\delta_{aver} = 12°$.

According to our results, it can be concluded that in a diffuser with $z_{dif} = 20$, the conversion (braking) of the gas flow velocity is less effective, due to the appearance of flow disruptions from the blade surfaces. At installation angle $a_3 = 20°$ and $a_4 = 53°$, the diffusivity of the inter-blade channel exceeds the recommended value by five times. The consequence of this is a sharp increase in total pressure losses due to vortex formation caused by changes in the flow structure in the channel. This, in turn, causes a decrease in the static pressure, which, at a given coefficient of hydraulic resistance of the network on which the compressor operates, causes a decrease in gas consumption and a corresponding deterioration in the thermal state of the turbogenerator.

Having assessed the above requirements and comparing the results obtained and the geometry of the diffuser channels, the following conclusions were drawn:

1. The use of a diffuser design with blades made according to the drawing in Fig. 1 with a total number $z_{dif} = 20$ compared to $z_{dif} = 28$ significantly worsens the efficiency of the blade diffuser in increasing the static pressure of the compressor under the same operating modes.

According to the constructed curve (Fig. 9), the change in the compressor head will be 12%. As for the geometry of the inter-blade channel, the length has decreased by 20%, and the cross-sectional area at the inlet of the channel has increased by 57%. Based on these data, we can say that the compressor pressure will decrease by 15–30% and will be about 690–840 mm H_2O (6770–8240 Pa), which is confirmed by the data of a three-dimensional simulation conducted in SolidWorks Flow Simulation.

2. In order to improve the parameters of the diffuser with $z_{dif} = 20$ and maintain the head at the level of the compressor with $z_{dif} = 28$, the following parameters were adopted:

– an increased radius of curvature of the blade profile to $R_{blade} = 1500$ mm;

– an increased blade length to $l = 387$ mm;

– a changed "direction" of the blade profile curvature, the swapped trough and back according to the sketch of the proposed new compressor diffuser;

– a provided gap between the diffuser and the compressor impeller at the level of 0.9 mm.

A sketch of the proposed structure of the new compressor diffuser with $z_{dif} = 20$ for the TGV-325 turbogenerator with a capacity of 325 MW is shown in Fig. 10.

3. To comply with the above parameters, it is necessary to:

– take the channel opening angle $\delta_{aver} = 20°$, which does not exceed, but is within the previously tested value $\delta_{aver} = 34°$ (at $z_{dif} = 28$); this provides the necessary compressor head, and thus improves the efficiency of the inter-blade channel;

– increase the channel length to 190 mm; while the overall dimensions $D_3 = 1316$ mm and $D_4 = 1670$ mm remain the same, and the recommended $l = 4 \cdot a = 4 \cdot 85 = 340$ mm will exceed the actual one by 1.8 times.

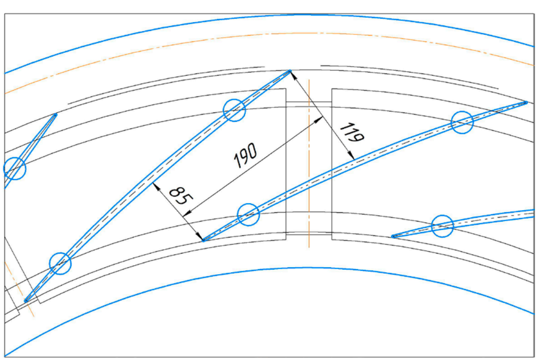

Fig. 10. Sketch of the proposed compressor diffuser structure with $z_{dif} = 20$

After changing the shape of the diffuser blades, the degree of expansion (diffusivity) of the inter-blade channel is ensured at the level of $b / a = 119 / 85 = 1.4$.

Thus, the shape of the inter-blade channel achieves significant similarity with the shape of the TGV-325 diffuser channel, and the compressor head remains at the level of compressors with the number of blades $z_{dif} = 28$ and the blade installation angle $a_3 = 20°$.

5. 3. The structure designed and the proposed manufacturing technology for the compressor diffuser power belt

We have designed the structure and proposed a manufacturing technology for the power belt of a multi-component compressor diffuser in the turbogenerator with a capacity of 325 MW. The designed structure makes it possible to reduce the influence of uneven loads on welded joints and fastening elements, which arises due to temperature gradients and uneven rigidity of the structure.

The general view of the proposed structure of the diffuser shield for the compressor in the turbogenerator TGV-325 is shown in Fig. 11. This structure meets the requirements for the electrical strength of the dielectric gas gap between the diffuser and the uninsulated end of the ventilation tube of the rod in the frontal parts of the stator winding in the turbogenerator.

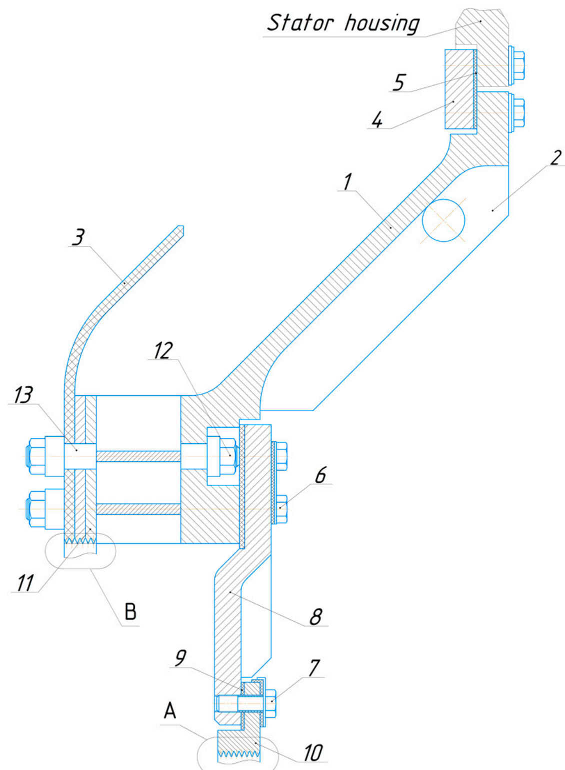

Fig. 11. Drawing of the proposed structure for a diffuser shield: 1 – main element of attachment of blades; 2 – additional rib; 3 – diffuser cover disk; 4 – shield attachment ring to the diffuser; 5 – fiberglass gasket; 6 – bolted connection; 7 – bolted connection; 8 – flange; 9 – fiberglass gasket; 10 – rotor seal; 11 – shield seal; 12 – bolted connection; 13 – fiberglass sleeve

The main element of the blade fastening is represented by element pos. 1 (Fig. 11), which is made of stainless steel of grade AISI 321. AISI 321 steel has high performance qualities and is characterized by resistance to corrosion and high temperatures. These properties enable the possibility of its use in aggressive environments. Due to the replacement of magnetic steel grade S235 according to the DIN EN 10025-2:2019 standard, which is used in many existing structures of diffusers for TGV-325 turbogenerators with AISI 321 steel in the proposed design, decarburization of steel is excluded.

The application of fiberglass as the basic material of the cover disk and AISI 321 steel as the main material of the blade fastening element makes it possible to significantly reduce circulating currents and overheating of the structure. This is achieved by breaking the current circuit (stator housing – diffuser shields through bolts or welded joints – guide blades and inner walls of the diffuser).

Due to the fact that the diffuser perceives loads caused by uneven deformation of the stator housing, and the gap between the impeller and the diffuser δ must be at the level of 0.9 mm, an additional rib pos. 2 was installed (Fig. 11). In this case, the weld of the rib with the main body of the structure must be of equal strength along the entire length. The necessary strength and corrosion-resistant properties of the weld can be achieved through the use of argon welding technology (ISO 4063:2009, ISO 15614-1:2017).

When welding stainless steel, it is necessary to take into account that when the seam is overheated, the material becomes excessively fusible. Also, when the temperature rises above 500°C, the product completely loses its resistance to corrosion. All this occurs due to chromium oxide, so it is worth avoiding overheating or cooling the part.

The easiest way to comply with temperature parameters is to reduce the current strength by 1/5 of the values that are suitable for steels of the same thickness without ligatures. It is also necessary to take into account the high coefficient of linear expansion. Metals tend to shrink, which provokes the appearance of cracks in the joint areas. To prevent this, it is necessary to observe the gaps specified for different materials in the regulatory documents for welding stainless steels (ISO 14343:2025, AWSD1.6/D1.6M:2017). It should be borne in mind that a high level of electrical resistance can lead to overheating not only of the part but also of the solder material. To summarize, during the welding process, it is necessary to carefully monitor the following criteria:

– temperature of the metal of the part;
– temperature of the solder;
– gap between the parts to be welded;
– current strength.

For the proposed diffuser structure, steel seals of the rotor pos. 10 (Fig. 11) and the shield pos. 11 (Fig. 11) were designed, which are shown in Fig. 12, 13, respectively.

The design of these elements includes three consecutive steps:

– the geometry of the seals forms;
– the resulting seals are attached to the flange pos. 8 (Fig. 11) made of non-magnetic steel using bolted connections pos. 6 and pos. 7;
– the flange-seal geometry is correlated.

With such a scheme, despite the large diameter of the main shield (due to the use of several force solutions of the bolted connections), damping of thermal displacements caused by deformation of the stator housing will be enabled.

In the proposed solution, the diffuser cover disk pos. 3 (Fig. 11) is made of fiberglass. To attach it to the main blade mounting element pos. 1 (Fig. 11), bolts made of stainless steel pos. 12 (Fig. 11) are used. In this case, two fiberglass bushings pos. 13 (Fig. 11) are used, which provide the necessary installation accuracy, which makes it possible to exclude additional deformations for the seals, shown in Fig. 13. The use of two fiberglass bushings provides the necessary installation accuracy, which makes it possible to exclude additional deformations for the seals. The seal on the rotor side is composite and consists of three segments: the first is fiberglass, and the rest are made of stainless steel.

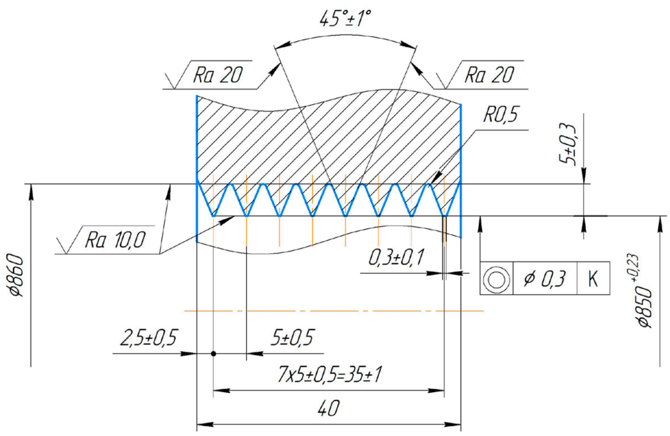

Fig. 12. Drawing of the steel rotor seal (view A, Fig. 11)

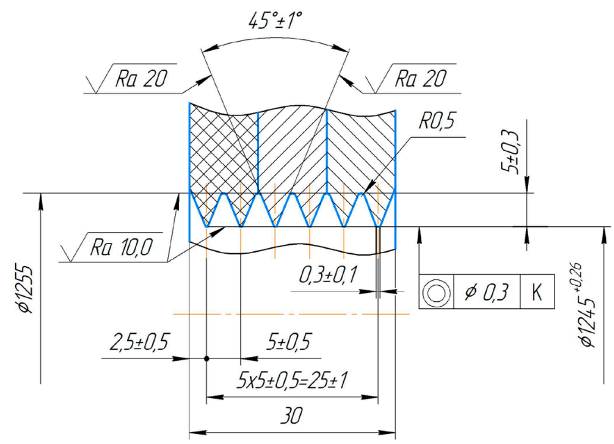

Fig. 13. Drawing of the steel seal of the shield (view B, Fig. 11)

6. Discussion of results based on investigating the proposed structures of the turbogenerator compressor diffuser and the diffuser power belt

The principal factor characterizing the efficiency of the compressor is the dependence of dynamic head on the mass flow rate of the cooling gas $H_{dyn} = f(Q_{gas})$. This dependence for the centrifugal compressor in the TGV-325 turbogenerator (Fig. 8) was established from the results of CFD simulation. The calculations took into account the parameters of the working fluid (hydrogen). Given that analysis of the results obtained by the finite volume method required determining the reliability of the resulting values, characteristic points were locally determined. To confirm them, tests of the centrifugal compressor were carried out at the manufacturer's bench. That allowed us to eliminate the shortcomings of works [6–9] through the availability of data from full-scale tests carried out at manufacturer's benches with injection units of various configurations. The difference in speeds according to the results of blowing by the CFD method and full-scale studies ranged from 0.5% to 1.2%. The calculation was carried out according to the hydrogen consumption parameters as the product of the speed and the area of the characteristic cross-section of the inter-blade channel.

The results of technical reports on the TGV-325 turbogenerators allow us to conclude that the diffuser blade installation angle $a_3 = 20°$ is the most effective when using the compressor to blow gas flow $Q_{gas} = 22$ m^3/s. This angle is most consistent with the angle of flow exit from the compressor impeller at high gas flows and provides maximum head. The dependence of static pressure of the compressor on the number of blades $H_{st} = f(z_{dif})$ at installation angle $a_3 = 20°$ with the same geometry of the blade profile is shown in Fig. 9. It demonstrates that with the number of blades $z_{dif} = 28$, the value of the cooling gas head reaches a local extremum.

An increase in the number of blades leads to an increase in the speed of the cooling gas, and as a result, to an increase in aerodynamic resistance (head drop).

Given the above, a decision was made to use a structure with 20 blades. Based on this solution, we further designed the geometry of the flow part with step-by-step adjustment of the geometric parameters to determine their optimal values. The resulting calculation data were confirmed by the results from tests conducted on operating samples. The purity of hydrogen during the tests was in the range of 97...98%. The above allowed us to eliminate the shortcomings of works [10, 11] by sequentially analyzing the head characteristics of the system with a small step increase in parameters with subsequent tests on operating samples. These parameters include head, number of blades, gap sizes, etc.

Our work reports a new concept for designing a turbogenerator compressor diffuser by determining the geometric parameters of its flow part.

The proposed structure has made it possible to solve the task of enabling the required flow rate and head of the cooling gas for the ventilation system in a high-power hydrogen-cooled turbogenerator. Our study has made it possible to design an effective multi-component structure of a diffuser compressor with the number of blades $z_{dif} = 20$ (Fig. 10). The shape of the inter-blade channel, in this case, achieves significant similarity with the shape of the diffuser channel in TGV-325 while the compressor head remains at the level of compressors with the number of blades $z_{dif} = 28$. The current study was performed using ASME PTC-10 methods taking into account the structure features and parameters of the working fluid of the cooling system (hydrogen) in a turbogenerator, which eliminated the shortcomings of work [12]. In our study, the correlation of design parameters along the entire structural path of the diffuser was introduced.

This work used classical gas dynamics equations. Reliability analysis of data from our calculation was correlated based on experimental data. That allowed us to quickly conduct a study on the possibility of reducing the number of blades to decrease the aerodynamic resistance of the flow part. Performing a study of the proposed compressor structure on a bench at the manufacturing enterprise with an excess hydrogen head of 0.3 MPa allowed us to obtain actual data. That has made it possible to eliminate the shortcomings of [13, 14] through the use of experimental values of the cooling gas parameters.

Our paper reports a new concept for designing a diffuser shield attachment, diffuser seals, their structural elements, as well as basic power components.

The study was conducted in two directions: reducing the mass of the structure and decreasing the effect of circulating currents. In world experience, these goals are achieved by separating the seals from the general structure of the shields, which have fairly high rigidity. However, this path has a significant drawback: to attach these elements, it is necessary to build additional frame structures, which worsen the cooling system and increase the mass of the turbogenerator.

Unlike the above, in our study it was possible to divide the diffuser structure into components. The first part was made of stainless steel with additional fins, and the second part (cover disk) was made of fiberglass (Fig. 11). This solution allowed us to proceed to the optimization of the flow part, which was carried out both computationally and experimentally.

The final stage of the work was the design of a seal structure made according to the "puff pie" principle, which includes fiberglass and metal non-magnetic components (Fig. 13). This solution has no analogs and allows for technologically accurate setting of seal position through the introduction of additional special bushings made of fiberglass pos. 13 (Fig. 11).

In the current study, the task of improving the thermal state of the turbogenerator elements was solved by replacing the cover disk material with fiberglass and introducing new insulating elements. The issue of power separation was resolved by reducing the mass through additional finning of the structure and designing multi-component assembly for power circuits. A separate seal with improved technological control capabilities was also introduced. The above has made it possible to eliminate the shortcomings of [15, 16].

Our study has made it possible to scientifically substantiate the effectiveness and feasibility of using the designed new structures of a multi-component diffuser for the turbogenerator compressor, mounting of the diffuser shields, diffuser seal, and their structural elements, as well as basic power components. Modern structure trends in mechanical engineering were taken into account, and the use of new materials was proposed. Existing methods of gas-dynamic calculations did not take into account the multi-component structure of hydrogen-cooled turbogenerator units.

The main feature of the proposed method is the combination of theoretical and experimental parameters, taking into account their complex mutual influence and step-by-step adjustment to determine the optimal geometric parameters.

The high adaptability of the devised method for designing diffusers using mathematical models based on the basic equations from gas dynamics with step-by-step structure optimization allowed it to be implemented in the workflow at LLC "KhEMZ" (Kharkiv, Ukraine).

The results of our study could be used while designing new medium and high-power turbogenerators, as well as when retrofitting existing ones, which involves replacing and upgrading the injection system components. For example, hydrogen-cooled turbogenerators with a capacity of up to 500 MW, which are the main generating capacities at many power plants in Ukraine.

This study has limitations regarding the application of the method to electric machines with a rotation frequency of 1500 rpm or 3000 rpm. Adaptation to other rotation frequencies requires additional research.

As an area of further research, it is planned to conduct a full-fledged three-dimensional analysis of the cooling system injection complex. The results of analysis could provide a scientific justification for the choice of new materials, provided that the necessary uniform strength and uniform stiffness of structures are ensured when designing the complex as a whole.

Further advancement of the proposed method involves the use of various criterion equations and methods for calculating the stress-strain state. This could make it possible to design the structures of multi-component diffusers for turbogenerator compressors, diffuser shield mounts, as well as their main power components, which would provide the necessary strength and cooling gas flow rates over an extended range of rotation frequencies.

7. Conclusions

1. We have drawn in detail the arrangement of blades on a diffuser for the TGV-325 turbogenerator with a capacity of 325 MW. According to the table of possible parameters for diffuser structures, which can be realistically implemented and have no technological limitations, a family of curves for the $H_{st}=f(Q_{gas})$ and $\eta=f(Q_{gas})$. dependences was constructed. Based on the gas dynamics equations and CFD simulation methods, a family of curves for dependences $H_{dyn}=f(Q_{gas})$ of the operating modes for the basic compressor in a turbogenerator was built. At that stage of our study, the emphasis was on the structural and technological layout of the diffuser assembly in terms of choosing the minimum gap.

It was found that the installation angle of diffuser blades $a_3=20°$ is the most effective when using the compressor to blow gas at flow rate $Q_{gas}=22$ m^3/s. This angle at high gas flows provides maximum head. According to the plot of the dependence of static head of the compressor on the number of blades $H_{st}=f(z_{dif})$ for installation angle $a_3=20°$, it was established that the value of the cooling gas head reaches a local extremum at $z_{dif}=28$.

In order to reduce the aerodynamic resistance of the flow part of the diffuser, the minimum number of blades $z_{dif}=20$ was determined, at which it is possible to provide the necessary head characteristics of the compressor by adjusting its basic geometric parameters.

2. In the course of analytical calculations using gas-dynamic equations, the use of a structure with 20 blades was scientifically substantiated. The compressor head remains at the level of compressors with the number of blades $z_{dif}=28$ and is $H_{st}=978$ mm H_2O.

It was found that at the angle of installation of the blades to prevent flow disruptions from blade surfaces $\alpha_3=20°$ and the number of blades $z_{dif}=20$, the appearance of flow disruptions from the blade surfaces is observed.

In order to improve the parameters of the diffuser with $z_{dif}=20$ and maintain the head at the level of the compressor with $z_{dif}=28$, the following parameters were adopted:

– an increased radius of curvature of the blade profile to $R_{blade}=1500$ mm;

– an increased blade length to $l=387$ mm;

– the changed "direction" of the blade profile curvature, the swapped trough and back according to the sketch of the proposed new compressor diffuser;

– the gap between the diffuser and the compressor impeller was provided at the level of 0.9 mm;

– the channel opening angle $\delta_{aver}=20°$ is adopted;

– the channel length is increased to 190 mm.

When the above conditions are met, the shape of the inter-blade channel achieves significant similarity with the shape of the TGV-325 diffuser channel. The compressor head remains at the level of compressors with the number of blades $z_{dif}=28$ and blade installation angle $a_3=20°$. An additional advantage of the proposed structure is the reduction in mass due to the smaller number of blades.

3. In our study, we managed to divide the diffuser structure into components. The first part was made of stainless steel with additional fins, and the second part (cover disk) was made of fiberglass. This solution makes it possible to optimize the flow part by reducing the gas dielectric gap between the cover disk and the frontal parts of stator winding.

It was proposed to replace the material of the main element of the blade fastening, which is magnetic steel S235, with non-magnetic AISI 321. Fiberglass was proposed as the main material of the cover disk and AISI 321 steel as the main material of the blade fastening element. Their use makes it possible to significantly reduce circulating currents and overheating of the structure due to the breaking of the current circuit (stator housing – diffuser shields through bolts or welded joints – guide blades and inner walls of the diffuser).

When designing the proposed structure, a method of composition of engineering alloys and synthetic materials, which have significantly different coefficients of thermal expansion, was used. In this case, the fastening of these elements was provided by stainless steel bolts and the additional inclusion of insulating sleeves in the structure, which made it possible to damp mechanical loads. To provide additional rigidity of the structure while reducing its mass, additional finning was carried out.

Recommendations have been offered on the technology of argon welding of stainless steel elements.

The final stage of our work was the design of a multilayer seal structure, which includes fiberglass and metal non-magnetic components. This solution allows for technologically accurate installation of the seal position through the introduction of additional special bushings made of fiberglass. This provides the opportunity to achieve the optimal value of the gap between the impeller and the diffuser δ at the level of 0.9 mm.

Conflicts of interest

The authors declare that they have no conflicts of interest in relation to the current study, including financial, personal, authorship, or any other, that could affect the study, as well as the results reported in this paper.

Funding

The study was conducted without financial support.

Data availability

All data are available, either in numerical or graphical form, in the main text of the manuscript.

Use of artificial intelligence

The authors confirm that they did not use artificial intelligence technologies when creating the current work.

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