

# On the rational form of rotary control diaphragms for steam turbines with industrial and cogeneration steam extraction

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**Abstract.** The Russian park of power steam turbines contains a large number of turbines with adjustable steam extraction from the flow path where rotary control diaphragms are used as flow controllers, which structurally easily fit into the flow path of these turbines. This method of regulating the steam flow rate is accompanied by a decrease in the efficiency of the subsequent stages of the turbine and causes the appearance, at reduced loads, of additional disturbing forces acting on the rotor blades. In the presented materials, a variant of a post-sampling stage with a radial rotary control diaphragm is considered. The performed mathematical modeling of the working fluid flows in such a stage showed that in this case, at all turbine loads, a relatively uniform velocity field is provided when steam enters the nozzle apparatus, which naturally entails the elimination of the noted drawbacks.

## 1 Introduction

In the Russian thermal power industry, along with condensing steam turbines, a large place is occupied by turbines with cogeneration and industrial steam extraction from the flow paths of these turbines.

This circumstance gives rise to a number of new requirements, both for the formation of their flow paths and for the modes of their operation.

In particular, the central requirement is the condition of the turbine operation independence both in terms of the heating or industrial steam extraction schedule, and in terms of the electrical load schedule.

This condition entails the need to vary the steam flow rate through the turbine cylinders in a very wide range.

If the pre-selection stages of the considered turbines operate mainly in about a fifty percent range of variation in steam flow rates, then in the post-selection stages this range reaches 90-95%. Accordingly, the regulating body providing such a wide range of operating modes of the post-extraction stages inevitably generates a very large unevenness of the velocity field in front of the inlet section of the nozzle apparatus of the subsequent stage,

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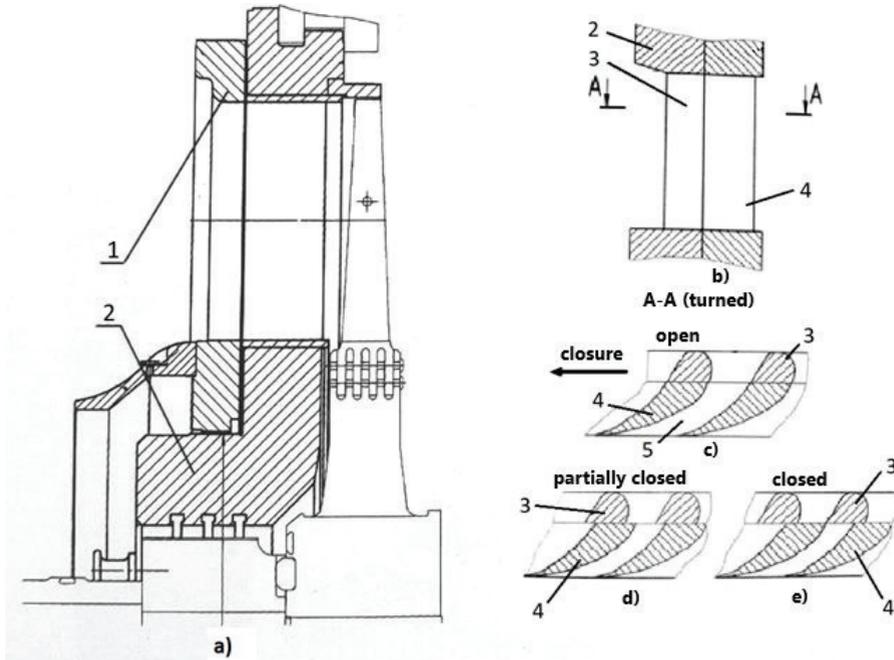
thereby greatly reducing the vibration reliability of the working blades of this stage. The specified working body for regulating the costs is in a very difficult situation from the point of view of ensuring the reliability of post-selection stages.

Here, in the final analysis, the central problem is the problem of providing a uniform velocity field in the mode of reduced loads when steam enters the nozzle apparatus of the post-extraction stage.

The existing bodies for regulating steam extraction from the flow path of the indicated turbines do not solve this problem. In this regard, it seems appropriate to consider the existing design solutions and consider possible ways to maintain uniform velocity fields in front of the working blades of the post-extraction stage with reduced steam consumption through such stages.

## 2 Typical rotary diaphragms of steam turbines with adjustable steam extraction and their disadvantages

In steam turbines with industrial and cogeneration steam extraction from their flow paths, axial rotary diaphragms are widely used as a regulator of these extractions. A typical design of such diaphragms is shown in Fig. 1 [1].

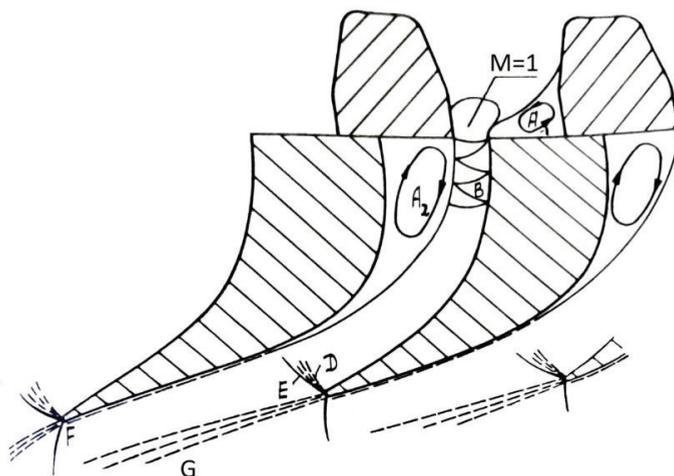


**Fig. 1.** The design of the PND rotary-regulating diaphragm. a, b - radial section, c, d, e - rotary-regulating diaphragm.

Structurally, the regulating diaphragm 1 is part of the nozzle apparatus 2 of the post-sampling stage, since in the open position, the profile baffles 3 of the diaphragm 1 are the inlet parts of the profiles of the nozzle 4. This solution provides an extremely low influence of the regulating body itself on the efficiency of the post-selection stage. The shape of the interscapular channel of the nozzle apparatus in this position of the control diaphragm is shown in Fig. 1c.

In aerodynamic terms, the reduced shape of the interscapular channel differs little from the shape of the interscapular channel in the lattice of profiles of conventional nozzles. According to the data of [2], the coefficient of profile losses of the considered lattice at a dimensionless velocity  $M_{1t} < 0,9$  does not exceed 4.5%.

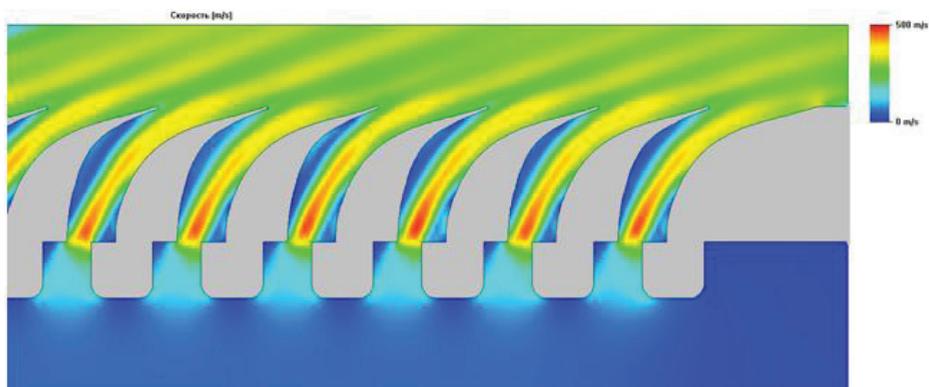
The situation changes dramatically when the diaphragm is turned and the inlet sections of the profiles are displaced to the position shown in Fig. 1d. The structural diagram of the flows in the channel formed in this way, when the diaphragm is opened by 25% [2] and the supercritical pressure drop across the nozzle apparatus is shown in Fig. 2.



**Fig. 2.** Scheme of vortex zones. A1, A2 - vortex zones, B - system of shock waves and rarefaction waves behind the adjustable section, D - rarefaction waves, E - internal edge shock, F - external edge shock, G - edge vortex wake

Such an overlap of the inlet section in the channel of the nozzle array entails an inevitable transition from stationary to unsteady flow with the formation of closed separation zones with intense vortex flow.

As a result, the rotor blades of the post-sampling stages perceive an unsteady flow with a highly irregular velocity field in the circumferential direction and low-frequency pulsations of velocities and pressures.



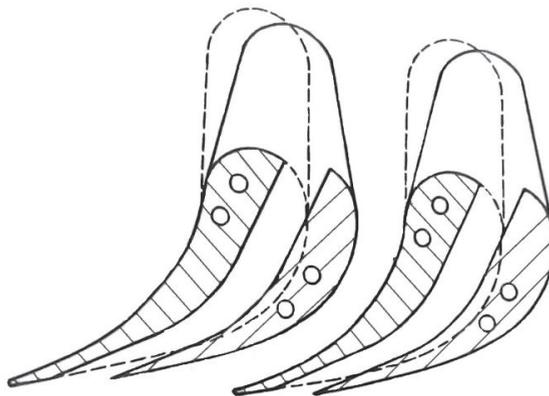
**Fig. 3.** Velocity field in the rotary diaphragm

The velocity field behind the nozzle apparatus with partial opening of the rotary diaphragm, shown in Fig. 3, obtained by mathematical modeling of flows, clearly confirms what has been said and agrees very well with the flow diagram shown in Fig. 2. If, to assess the degree of unevenness of the velocity field behind the nozzle apparatus of the post-selection stage, we use as the coefficient of unevenness  $\gamma_H$  the difference between the maximum velocity behind the nozzle apparatus  $C_{max}$  and the minimum velocity  $C_{min}$ , referred to the average velocity  $C_{cp}$  ( $\gamma_H = \frac{C_{max}-C_{min}}{C_{cp}}$ ), then when the rotary diaphragm is closed by 50%, the specified coefficient  $\gamma_H$  turns out to be equal to  $\gamma_H = 0,72$ . At the same time, according to experimental data [2], the profile losses in the annular lattice increase from 5% at full opening of the rotary diaphragm to 55%.

Naturally, with such a velocity field in the inlet section, the impeller blade unit experiences very large dynamic loads, which sharply reduce the vibration reliability of the post-sampling stages.

A very original way of improving the aerodynamic qualities of a nozzle apparatus with a rotary diaphragm is considered in [2], where not the inlet part of the nozzle lattice profiles was connected to the rotary diaphragm, but its entire part, which forms the convex surface of the profiles.

In this case, when the diaphragm is turned, the shape of the entire interscapular canal changes as shown in Fig. 4.



**Fig. 4.** Rotary diaphragm with variable channel shape

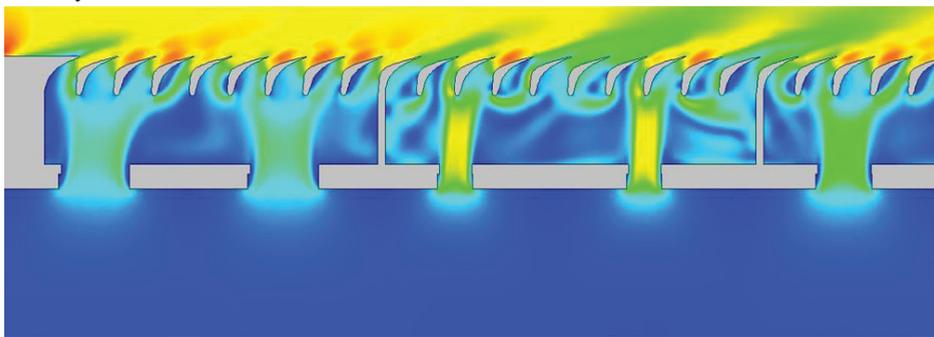
If in the original rotary diaphragm, when it is rotated, the steam inlet into the subsequent interscapular channel is carried out with a sudden expansion (Fig. 1), then in this case, when the passage area of the interscapular channel changes, a smooth bypass is maintained and, accordingly, the profile losses (compared to the original version) decreased by 15% (when closing by 50%) with a simultaneous decrease in the coefficient of unevenness of the velocity field.

However, due to the constructive complexity, such a solution turned out to be not rational.

The solution considered by A.A. Zhinov turns out to be simpler and, at first glance, quite reasonable. In the Kaluga branch of the Bauman Moscow State Technical University, where the rotary diaphragm is separated from the nozzle apparatus of the post-selection stage. At the same time, in order to reduce losses from throttling, the new rotary diaphragm provided the nozzle principle of throttling steam in a partial turbine operation mode. Here, with limited axial dimensions, the principle of nozzle steam distribution was implemented, which is characteristic of high-pressure cylinders of most powerful steam turbines. Accordingly, all the disadvantages inherent in this type of steam distribution [3] have remained, the main of which is the partial (in the circumferential direction) of the steam supply to the chamber between the nozzles of the regulating bodies (rotary diaphragm) and the steam inlet into the nozzle array of the post-extraction stages.

As shown in [3], even with relatively large axial dimensions of the said chamber, the subsequent stage perceives the flow with practically the same circumferential unevenness of the velocity field that is generated by the control stage. In this case, such a generator of circumferential irregularity is an axial rotary diaphragm.

In confirmation of the above, Fig. 5 shows the results of mathematical modeling of the velocity fields in the flow path of the entire complex under consideration, consisting of a rotary diaphragm, an equalization chamber and a nozzle apparatus, performed at the Kaluga branch of the M.V. Bauman. It is clearly seen here that, indeed, the unevenness of the velocities [2], which is generated by the axial rotary diaphragm, passes through the axial space of the alignment chamber without a noticeable decrease in the initial unevenness of the velocity field.



**Fig. 5.** Velocity field when steam enters the post-extraction stage (full opening of the control diaphragm)

Thus, the considered constructive solutions do not provide an increase in the efficiency and reliability of the blade apparatus of the post-selection stages, while maintaining the axial rotary diaphragms as a regulating body.

In aerodynamic terms, the problem under consideration can be solved by switching from axial to radial diaphragms, which were considered earlier in [4, 5, 6].

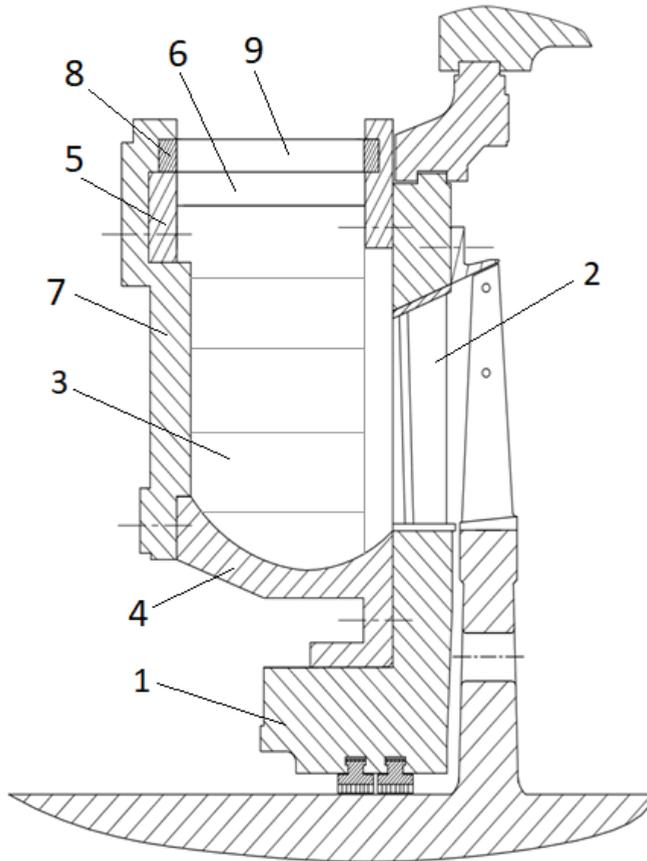
### 3 Constructive version of the radial rotary diaphragm

As an object for aerodynamic research by the method of mathematical modeling of the efficiency of using radial rotary diaphragms, a control radial rotary diaphragm was developed, inscribed in the flow path of the T-56/73-7,8/0,04 KTZ turbine, instead of the existing rotary axial diaphragm.

The structurally investigated rotary diaphragm connected with the nozzle apparatus of the post-sampling stage is shown in Fig. 6.

In this case, the steam supply to the nozzle apparatus 1 with a blade apparatus 2 comes from a special annular chamber 3, consisting of a lower annular profiled bypass 4 connected to the body of diaphragm 2, an upper annular bypass 5 with windows 6 for steam passage on the annular side surface of the bypass 5, and end cap 7. The minimum axial size of the chamber is determined from the conditions of equality of the total area  $F_{01}$ , windows 6.

A rotary radial diaphragm 8 with windows 9, counter windows 6 on the outer cylindrical surface of the chamber 3, were installed on this surface and fixed with annular grooves on the end surface of the cover 7 and in the upper bypass 5.



**Fig. 6.** Rotary diaphragm design

When the diaphragm 8 is turned, the windows 6 on the stationary cylinder 5 are overlapped by the walls of the diaphragm 8, thereby reducing the flow rate of steam supplied to the post-extraction stages. With the complete overlap of windows 6, a certain leakage of steam

remains through the inevitable gap between the cylindrical surface of the bypass of the chamber 3 and the cylindrical inner surface of the diaphragm 8. Since in a purely heating mode it is necessary to supply a certain amount of steam to the post-extraction compartment to cool its blade apparatus, in this case the leak steam through the inlet diaphragm is not a disadvantage.

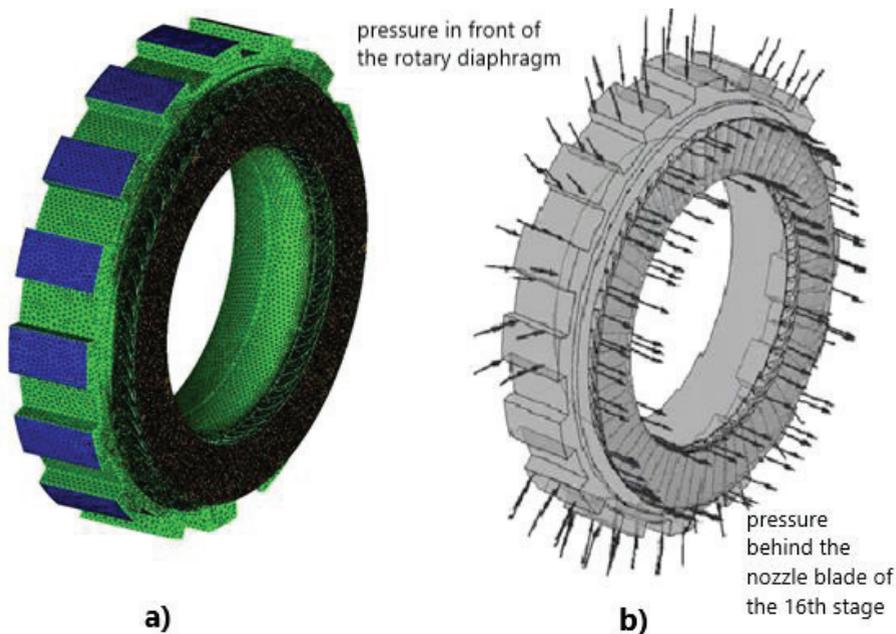
The proposed modernization of the executive bodies for regulating the steam flow rate in the cogeneration extraction of a particular turbine does not require changing the existing rotary diaphragm drive system.

In addition, when using a cylindrical rotary diaphragm, there is no need to unload it from axial forces, which is very important for turbines with industrial steam extraction.

The extent to which the proposed solution solves the problem of ensuring a uniform velocity field when steam enters the nozzle apparatus of the post-sampling stages can be judged from the results of the mathematical modeling carried out below for the steam flow in the flow part of the chamber 3 connected to the nozzle apparatus of the post-sampling stage.

#### 4 Methodology for the study of steam in the nozzle apparatus of the post-extraction stage using a radial rotary diaphragm and its results

To carry out numerical simulation of the rotary diaphragm together with the post-sampling stage, a 3D model of their vapor volume and a computational grid were built (Fig. 7a). To improve the accuracy of the calculation, the mesh was locally refined in the area of the nozzle apparatus, the passage windows of the diaphragm and the mixing chamber.



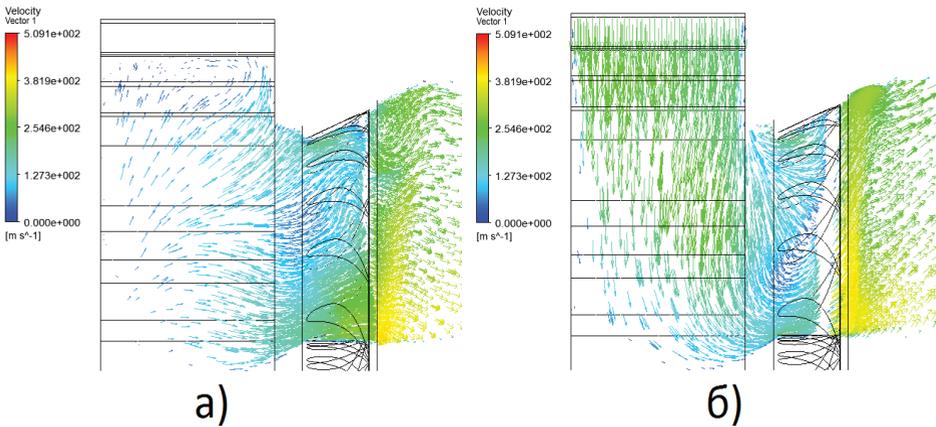
**Fig. 7.** Computational grid of the vapor volume of the rotary diaphragm and imposed boundary conditions

The boundary conditions imposed on the model (Fig. 7b) were the thermodynamic parameters of the steam in front of the rotary diaphragm and behind the nozzle apparatus of the first post-extraction stage of the turbine. The classical equations of conservation of mass, energy and momentum were taken as the calculation equations, and the k-ε model was used as the turbulence model.

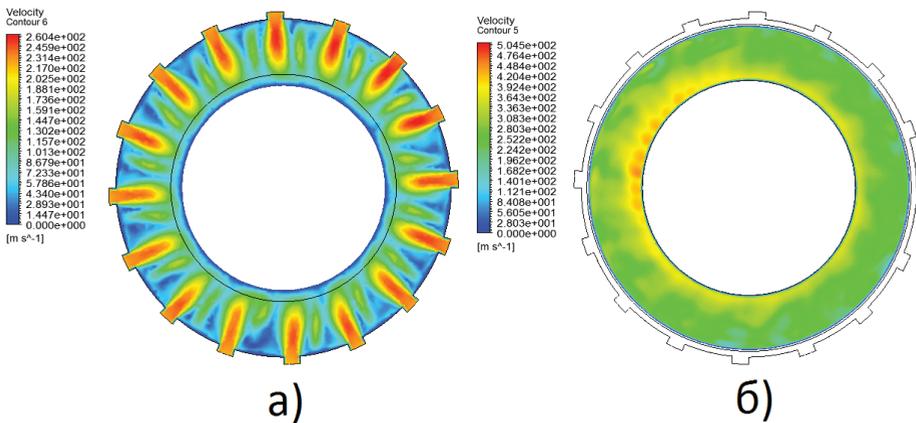
The results of mathematical modeling of the velocity fields in the meridional section of the rotary diaphragm in the condensation mode are shown in Fig. eight.

It is clearly seen here that with the full opening of the diaphragm, a practically uniform field of steam velocities at the entrance to the nozzle apparatus is actually provided, similar to the velocity field when using the original axial diaphragm.

However, the most interesting is the mode with partial overlap of supply windows 6 (Fig. 6)



**Fig. 8.** Velocity vector field a) in the meridional section passing between the entrance windows b) in the meridional section passing through the entrance window



**Fig. 9.** Velocity field in the inlet and outlet sections when the diaphragm is opened by 50%

For this particular case, Fig. 9 shows the flow pattern in the meridional section of the post-sampling stage with an attached control diaphragm and the end velocity field when entering the steam directly into the nozzle apparatus.

It is clearly seen that in chamber 3 there is indeed an almost uniform velocity field (Fig. 9a). The velocity field in the circumferential direction at the outlet of the nozzle apparatus also changes very little (Fig. 9b), and in the radial direction, due to an increase in the step between the profiles in the nozzle lattice, the velocities naturally decrease.

It is important to note that in the transition from a flat rotary diaphragm to the considered annular rotary diaphragm, the additional dynamic load on the impeller blades of the post-sampling stage practically disappears, due to the influence of the uneven flow of the impeller profiles of the stage impeller entering the array.

## 5 Conclusions

1. The analysis of the known methods of reducing the dynamic loads on the rotor blades of the post-extraction stages caused by the disturbances introduced into the steam flow by the regulating rotary diaphragms showed that these methods, while maintaining the axial rotary diaphragms, cannot solve this problem.
2. The carried out mathematical modeling of the flow in a complex consisting of a radial rotary diaphragm, a chamber for leveling the velocity field and a subsequent nozzle apparatus showed that with such a combination of a regulating body with a subsequent turbine stage, a uniform velocity field is provided in all turbine operation modes as when steam enters the nozzle apparatus of the post-extraction stage, and when the steam leaves it.
3. The flows obtained as a result of the conducted studies of the flows in the nozzle apparatus of the post-extraction stage with an upstream radial rotary diaphragm clearly show that when using these diaphragms, not only the problems of increasing the efficiency of the post-extraction compartments of turbines with industrial and cogeneration steam extraction are solved, at reduced costs, but also increased vibration reliability of the rotor blades of the entire post-sampling section of the turbine.

## References

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