energies

Article

Numerical Simulation of Gas Flow Passing through Slots of Various Shapes in Labyrinth Seals

Vadym Baha [1], Natalia Lishchenko [2], Serhiy Vanyeyev [1], Jana Mižáková [2,*], Tetiana Rodymchenko [1] and Ján Piteľ [2]

[1] Faculty of Technical Systems and Energy Efficient Technologies, Sumy State University, Rymskogo-Korsakova St. 2, 40007 Sumy, Ukraine; v.baga@kttf.sumdu.edu.ua (V.B.); s.vaneev@kttf.sumdu.edu.ua (S.V.); t.rodymchenko@kttf.sumdu.edu.ua (T.R.)
[2] Faculty of Manufacturing Technologies, Technical University of Kosice, Bayerova 1, 08001 Prešov, Slovakia; natalia.lishchenko@tuke.sk (N.L.); jan.pitel@tuke.sk (J.P.)
* Correspondence: jana.mizakova@tuke.sk; Tel.: +421-556026409

Abstract: Labyrinth seals are widely used in centrifugal compressors, turbines, and many other pneumatic systems due to their simplicity of design, reliability, and low cost. The calculation scheme for the movement of the working medium in a labyrinth seal is constructed by analogy with the movement of the working medium through holes with a sharp edge. Annular and flat slots, holes, and such a factor as the shaft rotation with a calculated sector of 3 degrees were studied. The purpose of the study is to determine the flow coefficient when the working medium flows through slots of various shapes. To achieve this purpose, modeling of the working medium flow in the FlowVision software was performed. The mass flow and flow coefficients are determined for the studied slot shapes. The convergence of the calculation results was determined by comparing the values of the mass flow rate at the inlet and outlet of the slot. Differences in visualizations of the flow for the studied variants of slots were established. The resulting difference should be taken into account in practical calculations of the working medium mass flow through the slot using a conditional flow rate factor which is determined by the slot design.

Keywords: labyrinth seal; flow coefficient; mass flow; slot; computational fluid dynamics method

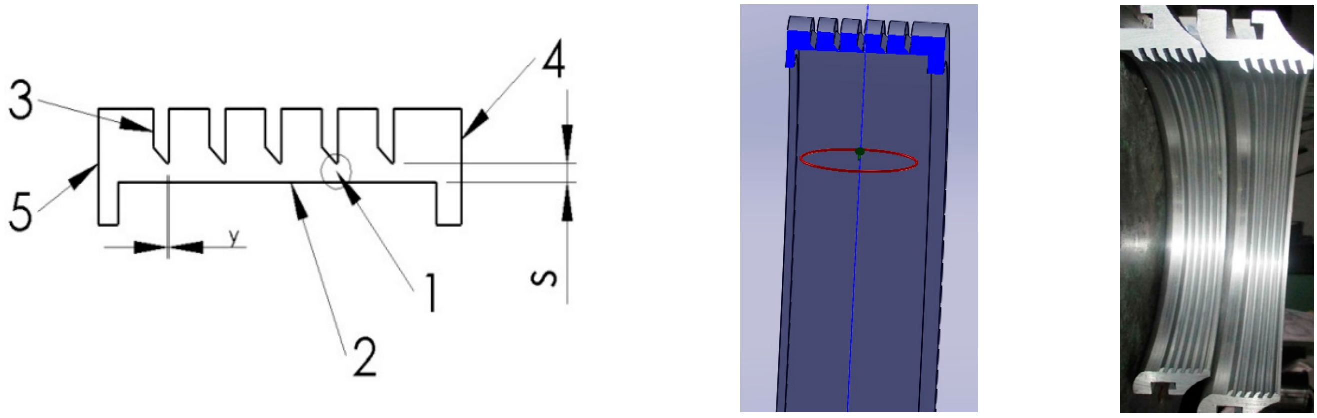
Citation: Baha, V.; Lishchenko, N.; Vanyeyev, S.; Mižáková, J.; Rodymchenko, T.; Piteľ, J. Numerical Simulation of Gas Flow Passing through Slots of Various Shapes in Labyrinth Seals. *Energies* **2022**, *15*, 2971. https://doi.org/10.3390/en15092971

Academic Editor: Dagmar Janacova

Received: 21 February 2022
Accepted: 14 April 2022
Published: 19 April 2022

1. Introduction and Related Works

Labyrinth seals are often used as internal seals of centrifugal compressors due to their simplicity and low cost. All internal seals of centrifugal machines operate with a guaranteed radial clearance (gap), which cannot be eliminated due to a number of technological reasons. The most economical method of improving the performance of any sealing device is to control its gaps. The given results in work [1] show the degree of influence of the value of radial clearances in the labyrinth seals of turbomachines on the amount of leakage. In addition to the gap, other geometric parameters that form the shape of the seal can also significantly affect the amount of leakage. In study [2], the results of the influence of geometric parameters on the operation of labyrinth seals are presented, which show the significance of the design of its microgeometry elements. More efficient seal designs are difficult to manufacture and may be unreliable in operation. To date, it can be argued that the limit of improving the flow path of a centrifugal compressor has been reached. However, the elements of sealing internal leaks are still a relatively large reserve for improving its efficiency, since the efficiency of turbomachinery depends on the magnitude of internal leakage through the labyrinth seals. Figure 1 shows the design of a labyrinth seal.



(**a**) (**b**) (**c**)

Figure 1. Labyrinth seal: (**a**) sketch of the flow path of the labyrinth seal, 1—slot (radial clearance s), 2—shaft, 3—sealing comb, 4—input section, 5—output section, y—sharp edge (y = 0.15 mm); (**b**) 3D model of the flow part of the labyrinth seal; (**c**) detachable labyrinth seal.

The question of the correctness of applying the known values of the flow coefficients to determine the mass flow of the working medium through the labyrinth seal remains relevant. The flow coefficient μ is defined as the ratio of the real mass flow to the theoretical one.

There are no theoretical studies of the complex flow mechanism in labyrinth seals; therefore, it is customary to use experimentally obtained flow coefficient μ for simplified schemes. For practical calculations of labyrinth seals, fundamental principles are used to determine the amount of leakage through a series of sequentially installed holes with sharp edges. They are supplemented by experimental coefficients obtained under conditions relatively far from real:

- the flat (not annular) slot models are often used,
- the effect of shaft rotation is not taken into account,
- the differences in slot geometry elements (the shape of a sharp edge, flow turbulence before and after the slot) are not taken into account, etc.

Therefore, in order to correctly determine the flow through labyrinth seals, taking into account the real shapes of the slots, it is necessary to perform a numerical simulation of this process.

An important issue in the design of labyrinth seals is the choice of a suitable design solution. These seals are widely used in a variety of technical devices. The article [3] provides a brief overview of the different types of seals already used in turbine construction and considers their adaptation to small turbomachines. It should be noted that the installation sites of labyrinth seals have a limited length, with axial movement of the rotor and stator parts. Since the calculation of mass flow through labyrinth seals is carried out by analogy with the calculation of mass flow through a hole with a sharp edge [4–6], it became necessary to numerically study the flow in slots of various geometric designs. Modeling allows to take into account the geometric elements forming a gap, edges as sharp as a blade, small gaps (s = 0.2 mm), shaft rotation, and the accuracy of the geometric position of the elements to obtain a visualization of the flow patterns. In sealing technology, it is necessary to organize the flow in such a way as to ensure complete washing out of the jet at the exit from the slot, increasing the dimensions of the device. The intensity of jet erosion is affected by a number of geometric and mode parameters.

Up-to-date software enables to obtain high-quality visualizations of a complex swirling flow in narrow slots and holes of any shape, to determine the flow rate through the slot, the values of density, temperature, pressure, etc. [7]. Flow visualization in sealing technology can directly point to those elements of the seal that need to be changed. So in study [6], the obtained flow pattern gave ideas for changing the design of labyrinth seals, which led to a significant increase in their efficiency. Screw seals and designs close to them are more effective after stepped seals. However, there are disadvantages such as clogging of the flow path, consumption of additional power, and high cost which have not made them widespread.

In study [8], a new labyrinth seal with a gear structure with stepped spiral teeth is proposed; the influence of geometric parameters on its operation was studied by the author using numerical simulation, which indicates the correctness of the use of Computational Fluid Dynamics (CFD) calculation for solving such problems. Additionally, in the works [9,10], attention is paid to labyrinth seals and their significant effect on the operation of turbine units. The works [4,11] are devoted to the study of the influence of geometric and regime parameters on the operation of labyrinth seals. The authors determined the flow coefficients of various studied models of slots designed without crucial simplifications. However, the flow coefficients obtained by them are correct only for the slot models which have been studied, not for general use.

In order to achieve a decrease in the slot flow coefficient μ, and consequently, the value of the mass flow through this slot m, it is necessary to increase the length of the flow jet run, or "force" the entire volume of the expansion chambers (the distance between the teeth) to work. To increase the sealing efficiency and to select the optimal slot shape (with the highest resistance to the working flow), it is necessary to conduct a series of systematic numerical experiments to obtain flow visualizations.

To determine the mass flow of the working medium through the labyrinth seal, A. Stodola proposed a calculation model consisting of sequentially installed holes with sharp edges instead of ridges (Figure 2). This analog model differs significantly from the design of a real seal: instead of an annular gap, a hole of the same section is taken; sharp edges occur only on one surface of the slot (ridges), while the other surface is formed by a smooth shaft. The operating conditions also differ significantly: the outflow from the hole implies an unlimited space at the output, and in the seal, after each slot, there is a limited chamber in which the kinetic energy may not be completely damped. Additionally, such a modeling method as creating a copy reduced by several times cannot be applied to labyrinth seals due to the presence of small axial clearances and a small ridge thickness.

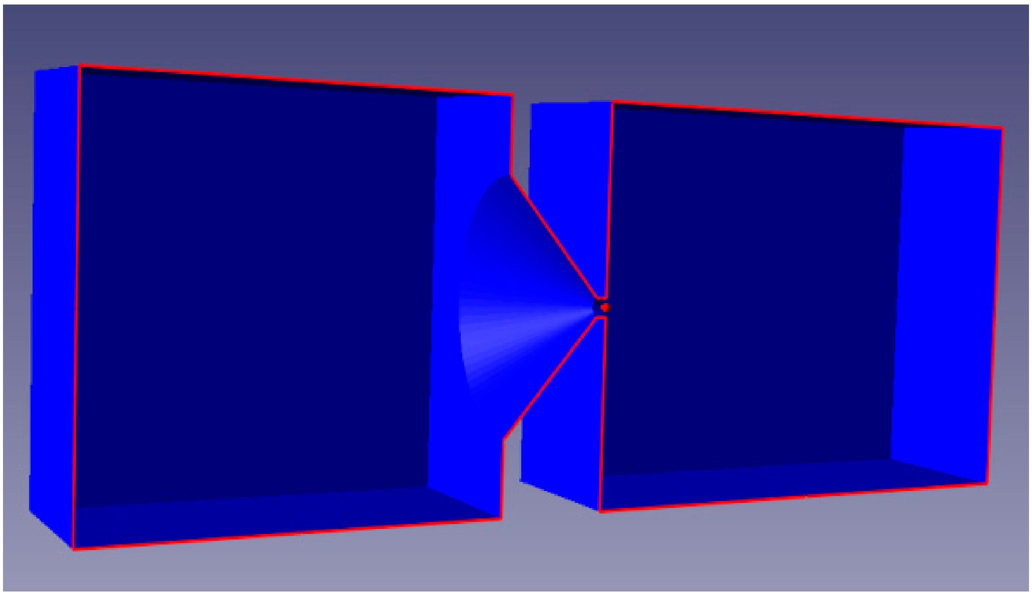


Figure 2. Calculation model of A. Stodola (hole with a sharp edge).

The key aspect of the need for further study of slots of simple shapes is that there are no theoretical studies of the complex flow mechanism in labyrinth seals. A. Stodola's formula [12] is often used for calculations as the classical formula for determining the flow through a hole with a sharp edge. It can be supplemented by experimental coefficients that take into account the type of seal, chamber dimensions, and tooth shape [13]. When calculating slots of various shapes, the concept of an equivalent slot is used, and then for the mass flow m through the area of the flow section f is valid:

$$m = \mu_p \cdot k \cdot f \sqrt{\frac{1 + (p_2/p_1)}{z}} \cdot \sqrt{(p_1 - p_2) \cdot \rho_1} \tag{1}$$

where μ_p is slot flow coefficient; k is coefficient factor determined from the Egley curves depending on the difference and the number of seal teeth; f is the area of the passage section of the slot (m^2); z is number of teeth seal; p_1 is seal input pressure (Pa); p_2 is seal output pressure (Pa); ρ_1 is seal input density (kg/m^3).

However, data obtained by simulation of the slot only with a flat shape using Formula (1) without considering the shape of the annular gap, shaft rotation, and the presence of flow swirling at the input is not valid enough. Such an empirical approach to the calculation of leakage through labyrinth seals has been used up to the present. Systematic studies of the working process of such seals have not been conducted. There is no rigorous methodology for modeling seal performance so far. Meanwhile, existing software (for example, FlowVision, Ansys CFX, APM FGA) makes it possible to study more detailed flows in channels of complex shapes. Taking these facts into account, the purpose of the study is to determine the flow coefficient when the working medium flows through slots of various shapes. To achieve this objective, modeling of the working medium flow in the FlowVision software was performed. The models of the labyrinth seal were taken from real labyrinth seals; 3D models of the flow paths of the studied slot geometries were built, then numerical calculations of the flow coefficient of these models were carried out in order to determine the value of the mass flow passing through these slot models.

The sections of this article are structured in the following manner. Following the introduction and related works in this section, a basic modeling and simulation methodology of gas flow passing through slots of various shapes in labyrinth seals is introduced in Section 2. Obtained results are presented in Section 3 including their discussion. Finally, Section 4 presents a summary of the article.

2. Materials and Methods

For numerical simulation, the FlowVision software package was used, which effectively solves a number of practical problems, including modeling the flow of gases and liquids in the flow parts of turbines, compressors, pumps, etc. This software environment uses a finite-volume approach to approximate the equations of a mathematical model.

Generally, the aim of fluid and gas modeling in the computational area is to obtain the characteristics of velocity, pressure, and other physical parameters of the fluid (gas). The calculation of these parameters must include the physical laws of their change, which comprehensively mathematically model the solved problem. A mathematical model of fluid or gas dynamics is a set of partial differential equations that determine the conservation laws (energy, mass, momentum) and fluid (gas) equations of state. This model describes the unstable/stationary motion of a mixture of two ideal gases at arbitrary Mach numbers.

FlowVision is based on the numerical solution of three-dimensional stationary and non-stationary equations of fluid and gas dynamics, which include the following laws: of conservation of mass, momentum (Navier–Stokes equations), equations of state:

- the Navier–Stokes equations:

$$\frac{\partial \boldsymbol{V}}{\partial t} + \nabla(\rho \boldsymbol{V} \otimes \boldsymbol{V}) = -\nabla P + \nabla\Big((\mu + \mu_t)\Big(\nabla \boldsymbol{V} + (\nabla \boldsymbol{V})^T\Big)\Big) + S \tag{2}$$

$$\frac{\partial \rho}{\partial t} + \rho \nabla \boldsymbol{V} = 0 \tag{3}$$

where $\boldsymbol{V}$ is velocity vector, P is pressure, μ is molecular dynamic viscosity, μ_t is turbulence dynamic viscosity, ρ is density, and S is as follows:

$$S = \left(\rho - \rho_{hyd}\right)\boldsymbol{g} + \rho\boldsymbol{B} + \boldsymbol{R} \tag{4}$$

where ρ_{hyd} is hydrostatic density, $\boldsymbol{g}$ is dispersion of mass concentration of working medium, $\boldsymbol{B}$ is rotation force (Coriolis, centrifugal), $\boldsymbol{R}$ is resistance force.

- The law of conservation of energy:

$$\frac{\partial(\rho h)}{\partial t} + \nabla(\rho V h) = \nabla\left(\left(\frac{\lambda}{C_p} + \frac{\mu_t}{Pr_t}\right)\nabla h\right) + Q \tag{5}$$

where h is enthalpy, λ is coefficient of thermal conductivity, C_p is specific heat capacity, Pr_t is turbulent Prandtl number, Q is heat energy.

We must also include the gap model in the solved problem. It is designed to take into account the resistance created by the narrow channel. The model is applied only in gap cells. When calculating the Navier–Stokes equations in the gap cells, the numerical approximation is adjusted, with the friction force being defined in this form:

$$F = -\rho \cdot D \cdot V, \tag{6}$$

where ρ is the flow density, V is the average flow velocity, and

$$D = 6\frac{\mu}{\rho s^2} \tag{7}$$

is a standard gap model, where s is gap size and μ is the coefficient of dynamic viscosity.

When turbulence models are used [14], the following conditions must be added to the list of boundary conditions for velocity on the impenetrable wall as follows.

Wall:

$$V_n = 0,\ \tau_w = \mu\left.\frac{\partial U}{\partial y}\right|_{V=0} \tag{8}$$

Tangential twist:

$$\omega,\ \tau_w = \mu\left.\frac{\partial U}{\partial y}\right|_{V=0} \tag{9}$$

Revolving wall:

$$V_n = 0,\ \tau_w = \mu\left.\frac{\partial U}{\partial y}\right|_{V=0} \tag{10}$$

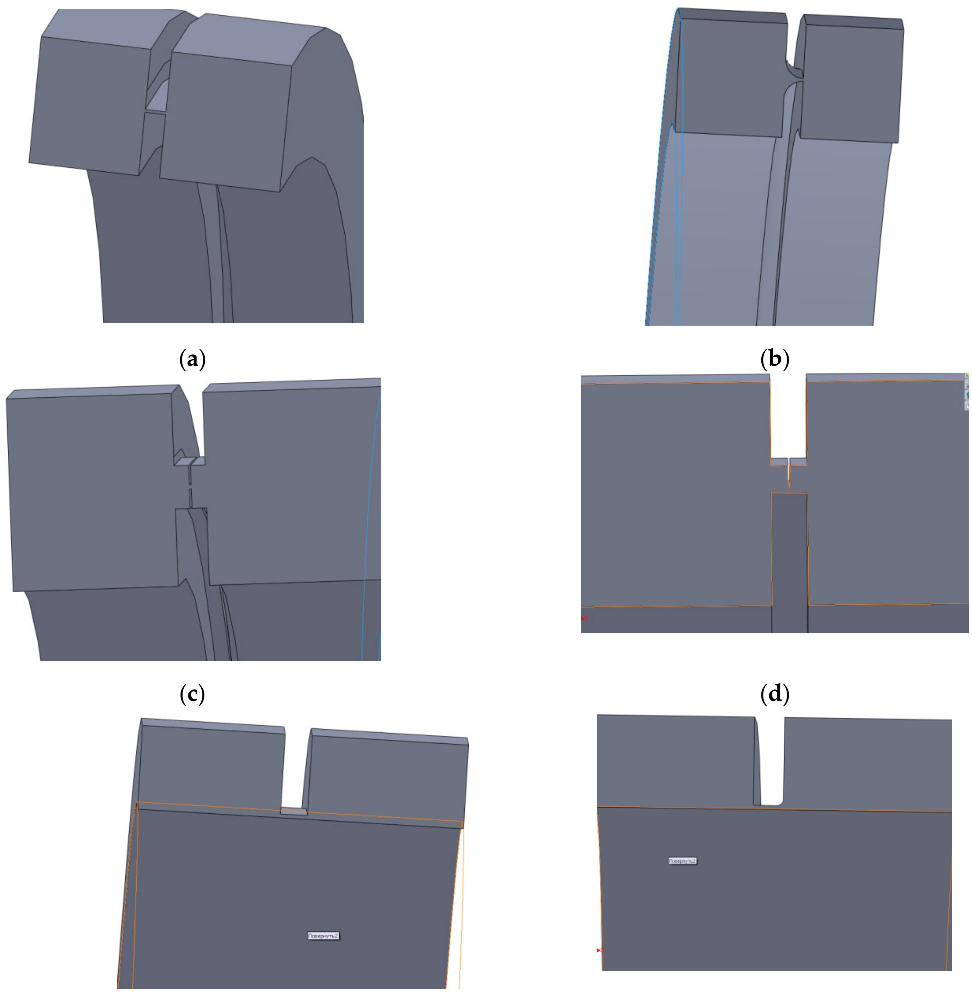
Under these conditions, it is necessary to set the equivalent roughness > 0 if the wall is known to be non-smooth. Special boundary conditions (Symmetry, Period, Slip) for k, ε, ω, υ_t are set in the same way as for other scalar variables. Experimental studies to determine the flow coefficient were carried out on a universal experimental stand [15], as close as possible to field conditions, and they were verified in [6].

The technique of numerical simulation of the flow considered the constructive shapes of the slots, as well as the rotation of the shaft. In the axisymmetric formulation, the sector of the flow path $\gamma = 3°$ was considered (when constructing the computational 3D model, the rotation of the sector, and not the entire circle, was used). The most suitable number of calculated cells lies within 10–16,000, which corresponds to 7–10 whole cells in the gap of the slot under study. The grid is built by selecting the required number of cells in the gap, and the number of calculated cells is automatically determined by the program and is an indicator of the density of the grid. The computational grid was built in such a way that about 10 cells were placed in the radial gap $s = 0.225$ mm, and so the number of calculated cells lies in the range of 10–16,000. The final computational grid consisted of

300–500,000 computational cells, depending on the size of the slot. A standard turbulence model $k - \varepsilon$, typical for this problem [2,3,5,6,8,13] was chosen.

For the chosen model, a fully compressible fluid, the time step, was first set equal to one tenth of the flight time of a particle of the working medium. The time of flight is the time of passage of an elementary particle of the flow from the inlet section of the model to the outlet section for 1 iteration. The simulation of the flow was carried out in a stationary setting; the air was considered as a perfect gas. The surfaces of the walls of the flow part were assumed to be hydraulically smooth. The values of static pressure and temperature were taken as input boundary conditions. The absence of large oscillations in the values of input and output parameters was taken as a convergence criterion, which is consistent with work [16].

To determine the correctness of considering the labyrinth seal of a centrifugal compressor as a series of holes installed in series, a number of calculations were made for slots with different geometry, with air as a working medium. The issues of rotor dynamics and bearing loads were not considered. Similar studies were carried out in [17,18]. The variants of slots under study are shown in Figure 3. When modeling the flow through slots of various shapes, chambers are made to ensure normalization of the flow before and after the slot.



(**a**) (**b**) (**c**) (**d**) (**e**) (**f**)

Figure 3. *Cont*.

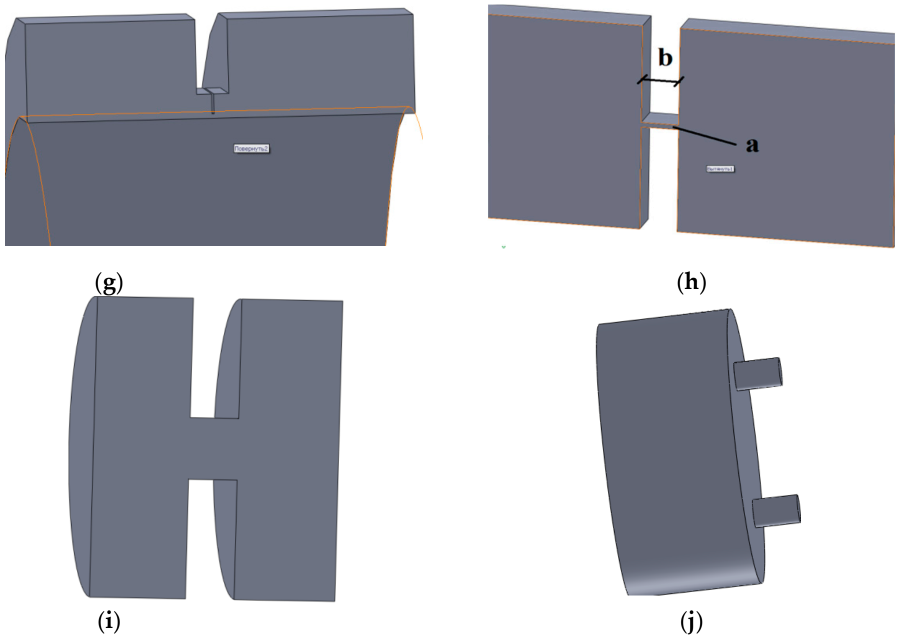


Figure 3. Seal models: (**a**–**g**) annular gap D = 240 mm, s = 0.225 mm; (**b**) r = 0.35 mm, 0.70 mm, and 1.59 mm; (**h**) flat slot (a = 0.225 mm, b = 1.6 mm); (**i**) hole (d = 2 mm, l = 1.6 mm); (**j**) two holes with d = 1 mm.

3. Results

Figure 4 below shows the visualization of the airflow for the considered variants of slots, as well as for (a) on Figure 3, including flow pattern (L) in Figure 4 for model (a) in Figure 3 which has a shaft rotation factor, from which significant differences in the flow pattern are visible.

Model (a) in Figure 3 was the base for comparing the flow pattern in annular slots, which differ in the degree of slot edge rounding (b) in Figure 3. As can be seen from the visualizations in Figure 4A–D, there are no qualitative changes, despite the increase in consumption characteristics from 0.0264 kg/s to 0.0384 kg/s, respectively (Table 1). It can be also seen, that under equal conditions, there is a different direction of jet exit in Figure 4A–E,G,H and for models (a), (b), (e), (f) in Figure 3.

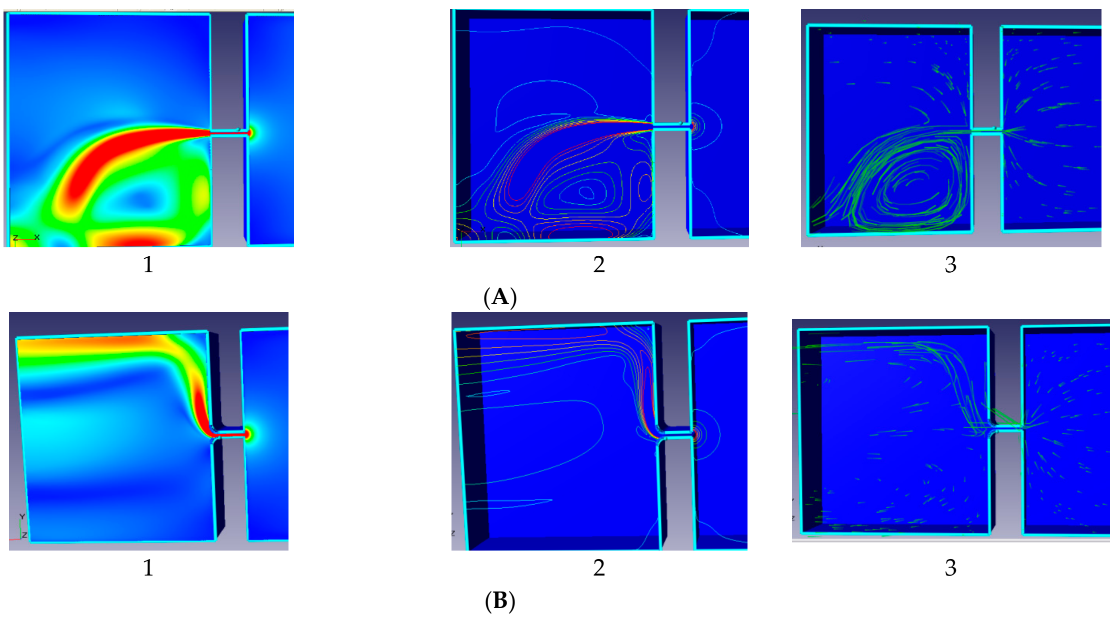


Figure 4. *Cont.*

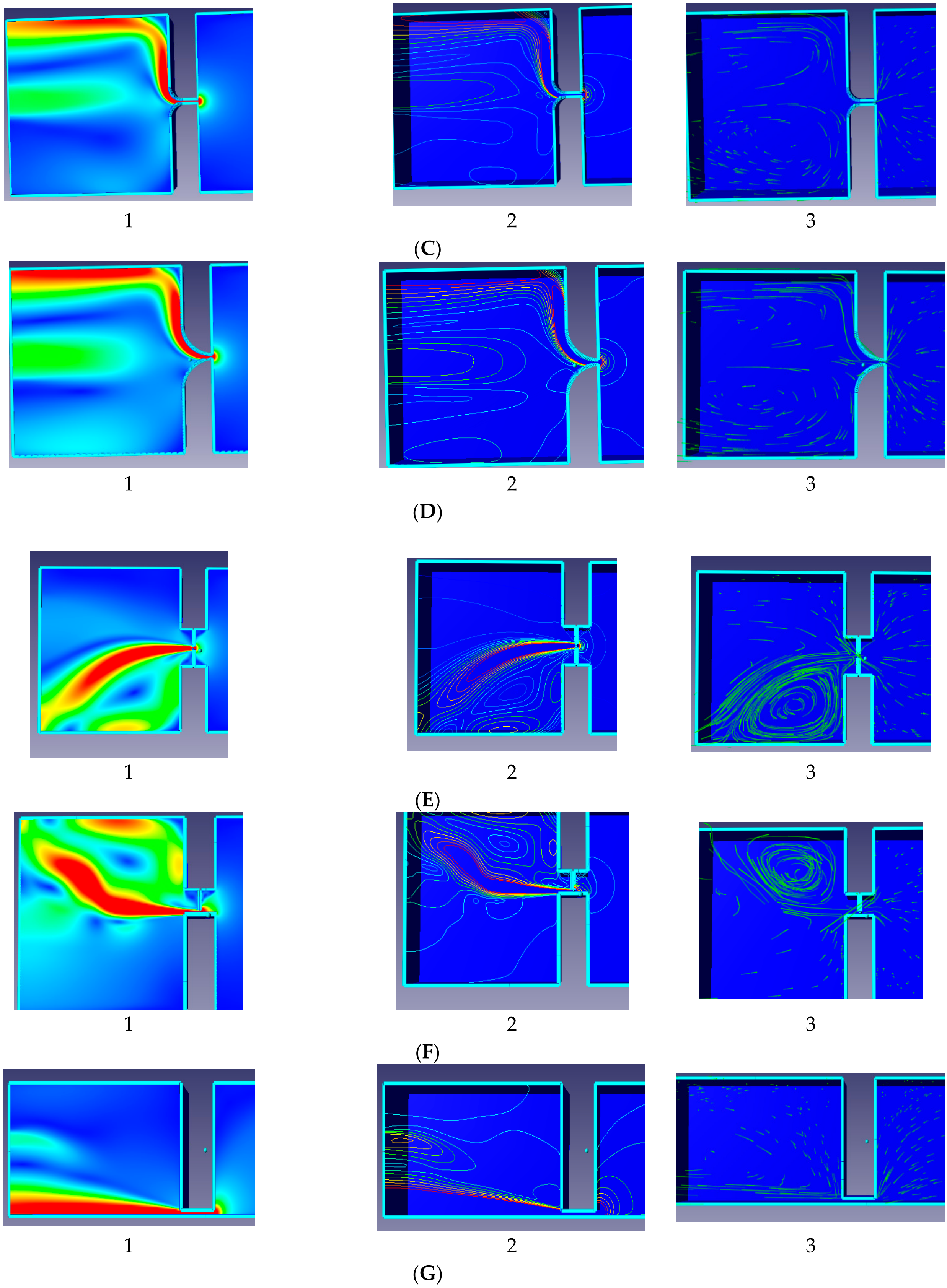
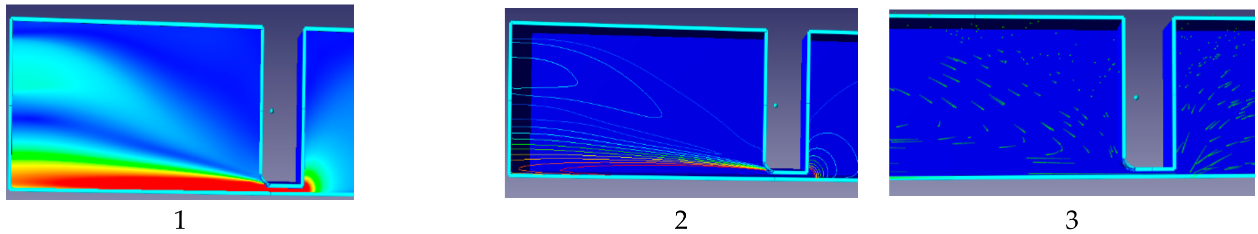
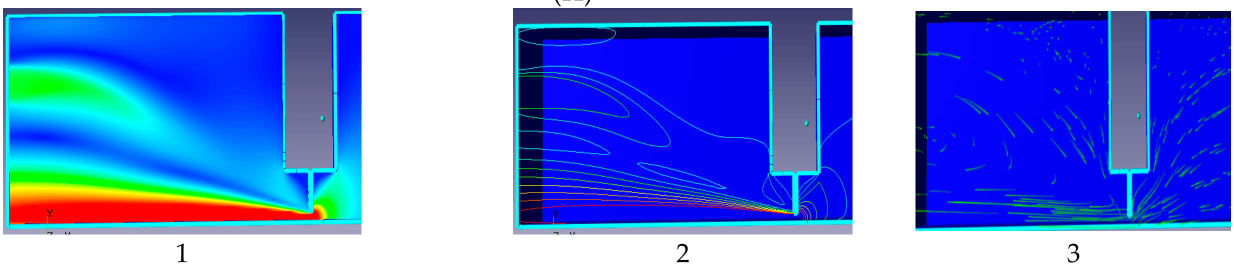


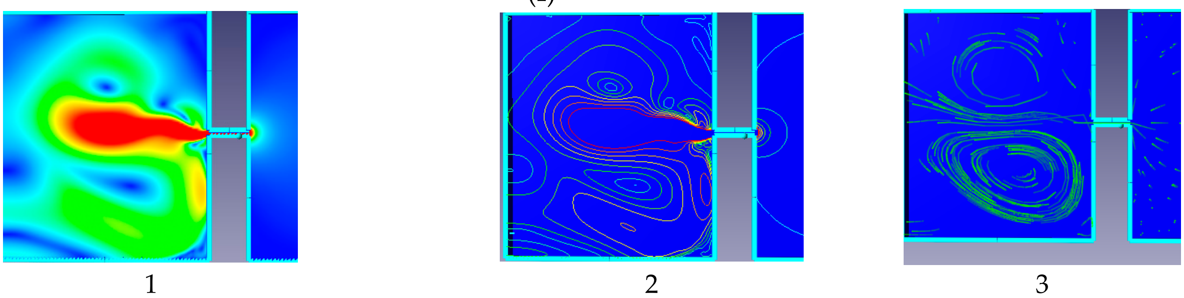
Figure 4. *Cont.*



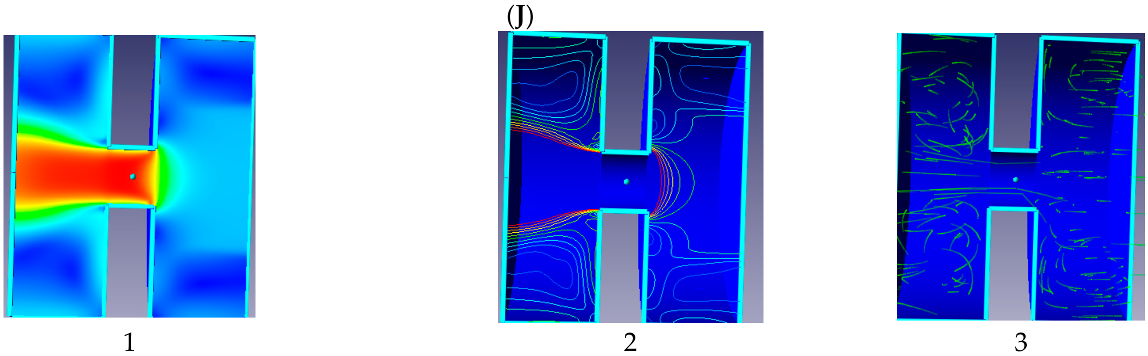
(H)



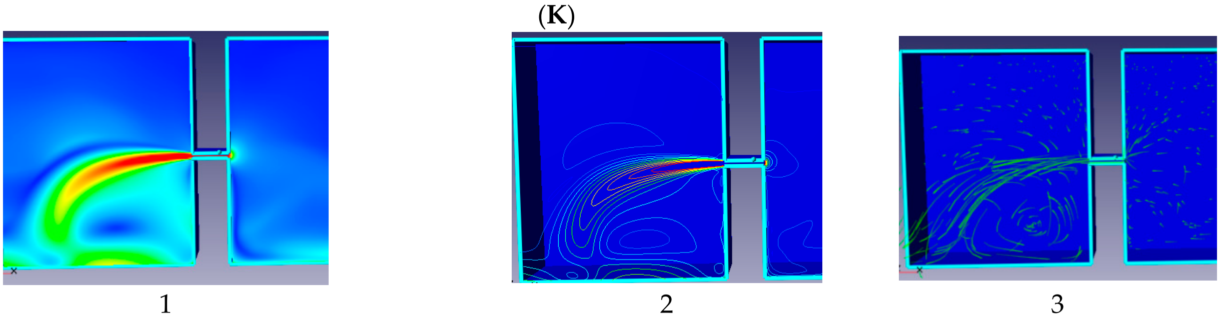
(I)



(J)

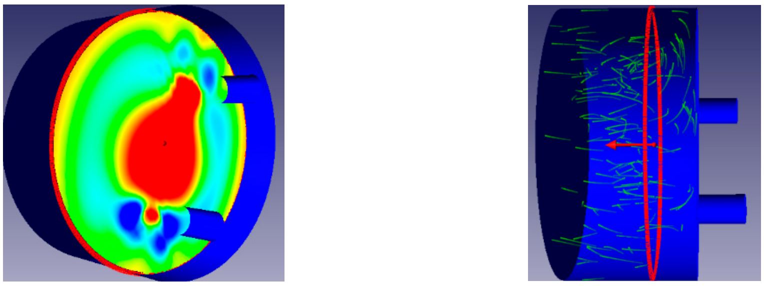


(K)



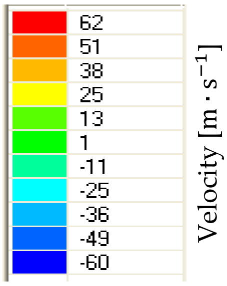
(L)

Figure 4. *Cont.*



1 2

(**M**)



Rainbow scale

(**N**)

Figure 4. Visualization of the flow pattern through slots of various shapes for slot models in Figure 3: (**A**) for model (a); (**B**) for model (b), r = 0.35 mm; (**C**) for model (b), r = 0.70 mm; (**D**) for model (b), r = 1.59 mm; (**E**) for model (c); (**F**) for model (d); (**G**) for model (e); (**H**) for model (f); (**I**) for model (g); (**J**) for model (h); (**K**) for model (i); (**L**) for model (a) with shaft rotation; 1—filling from a modulus of velocity, 2—modulus of velocity isolines, 3—flash out of velocity; (**M**) for model (j); (**N**) Rainbow scale.

Table 1. Results of a numerical study $\left(\frac{p_1}{p_2} = 1.4\right)$.

Model (Figure 3)	m (kg/s)	μ (−)
(a)	0.03	0.716
(b)		
r = 0.35 mm	0.0264	0.600
r = 0.70 mm	0.0294	0.702
r = 1.59 mm	0.384	0.916
(c)	0.0252	0.601
(d)	0.0348	0.831
(e)	0.0286	0.683
(f)	0.0344	0.821
(g)	0.0270	0.647
(h)	0.000067	0.754
(i)	0.0006	0.777
(j)	0.0007	0.907

Slots formed by one hole with a diameter 2 mm in model (i) in Figure 3 and two holes with a diameter 1 mm each in model (j) with equal area in Figure 3 were studied and the corresponding patterns (K) and (M) were obtained. It turned out that they have a different flow rate (m and μ in Table 1). The difference in the flow coefficient is 14%, which is explained by the presence of individual design features of each slot, which must be considered. Flow patterns (A), (E), (F), (J), (K) in Figure 4 have a formed vortex zone at the output due to the presence of free space in the flow part of the slots. As the number of teeth increases, the flow pattern changes.

The effect of rotation of the inner boundary of the annular chamber according to model (a) in Figure 3 is shown by pattern (L) in Figure 4. It leads to a decrease in the flow coefficient through the slot by 12% ($m = 0.0264$ kg/s, $\mu = 0.630$), which indicates the need to consider the rotation of the shaft in labyrinth seals.

From the foregoing, it follows that there is incorrectness in replacing real labyrinth seals with slots of different shapes by analogs with holes of the equivalent area due to the presence of significant differences in the flow patterns of the considered slots.

The results of calculations in the FlowVision software are shown in Table 1. The mass flow m was determined by the FlowVision. The flow coefficient μ was determined as the ratio of the obtained mass flow (Table 1) to the theoretical one (Formula 1).

It can be seen from Table 1 that all slot options under study have a different flow coefficient μ. The physical pattern of gas flow in round holes (model of A. Stodola), as well as in annular and flat equivalent slots, differs significantly. Differences in flow coefficient (Table 1) exceed 30%. Such large discrepancies indicate a significant influence of the geometry of the slot flow path on its flow characteristics. This implies the need for an individual approach when creating calculation methods for determining the flow characteristics of slots.

4. Conclusions

1. The use of the CFD software made it possible to perform a series of calculations of gas flow in narrow annular, flat slots, and holes, which made it possible to compare their flow characteristics and obtain flow visualizations.
2. The flow coefficients of slots of various shapes and holes with an equivalent area of the passage section were determined, which indicated the incorrectness of replacing real slot designs with holes with an equivalent cross-sectional area. To obtain accurate seal characteristics (flow factors), the individual characteristics of each gap must be considered.
3. The considered models of slots and holes with the same equivalent areas have different flow coefficients. The differences exceed 30%, which is significant and, in turn, leads to inaccuracies in the calculation of seals. The obtained discrepancies must be considered in practical calculations of the amount of leakage through the gap by introducing a conditional flow rate factor.
4. It has been established that there is a different direction of jet exit for patterns (A) and (E) in Figure 4. By changing the jet direction, it is possible to increase its range, forcing a larger volume of the expansion chamber to work. It leads to a decrease in the value of the slot flow coefficient and consequently, efficiency increases.
5. The effect of rotation of the inner boundary of the annular chamber leads to a decrease in the flow rate of leakage through the gap by 12%, which indicates the need to take into account the rotation of the shaft in labyrinth seals.

Author Contributions: Conceptualization, V.B.; methodology, V.B. and T.R.; software, V.B. and S.V.; validation, V.B. and J.P.; formal analysis, V.B., S.V. and N.L.; investigation, V.B. and T.R.; resources, J.P.; data curation, V.B. and J.M.; writing—original draft preparation, V.B.; writing—review and editing, V.B., N.L. and T.R.; visualization, V.B., N.L. and J.M.; supervision, V.B., J.P. and N.L. All authors have read and agreed to the published version of the manuscript.

Funding: This work was supported by the Slovak Research and Development Agency under the contract number APVV-15-0602 and also by the Ministry of Education, Science, Research and Sport of the Slovak Republic within the research project VEGA 1/0704/22.

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: The data presented in this study are available on request from the corresponding author.

Conflicts of Interest: The authors declare no conflict of interest.

References

1. Childs, D.W. *Turbomachinery Rotordynamics with Case Studies*; Minter Spring Publishing: Minter Spring, TX, USA, 2016.
2. Zhang, W.; Wu, K.; Gu, C.; Tian, H.; Zhang, X.; Li, C. Swirl Brakes Optimization for Rotordynamic Performance Improvement of Labyrinth Seals Using Computational Fluid Dynamics Method. *Tribol. Int.* **2021**, *159*, 106990. [CrossRef]
3. Zaniewski, D.; Klimaszewski, P.; Klonowicz, P.; Lampart, P.; Witanowski, Ł.; Jędrzejewski, Ł.; Suchocki, T.; Antczak, Ł. Performance of the Honeycomb Type Sealings in Organic Vapour Microturbines. *Energy* **2021**, *226*, 120242. [CrossRef]
4. Zhang, B.; Dou, Y.; Hong, Q.; Ji, H. Experimental Investigation of Effect of Tooth Geometrical Parameters to Flow Characteristics in Slant Labyrinth Seals. *Concurr. Comput. Pract. Exp.* **2018**, *30*, 24. [CrossRef]
5. Alizadeh, M.; Nikkhahi, B.; Farahani, A.S.; Fathi, A. Numerical Study on the Effect of Geometrical Parameters on the Labyrinth–honeycomb Seal Performance. *Proc. Inst. Mech. Eng. Part G J. Aerosp. Eng.* **2018**, *232*, 362–373. [CrossRef]

6. Bondarenko, G.; Vanyeyev, S.; Baga, V.; Rodymchenko, T.; Bashlak, I. Increase of Efficiency of Turbine Setting Based on Study of Internal Flows. In *Advances in Design, Simulation and Manufacturing*; Springer: Cham, Switzerland, 2018; pp. 237–246.
7. Khovanskyi, S.; Pavlenko, I.; Pitel, J.; Mizakova, J.; Ochowiak, M.; Grechka, I. Solving the Coupled Aerodynamic and Thermal Problem for Modeling the Air Distribution Devices with Perforated Plates. *Energies* **2019**, *12*, 3488. [CrossRef]
8. Zhou, W.; Zhao, Z.; Wang, Y.; Shi, J.; Gan, B.; Li, B.; Qiu, N. Research on Leakage Performance and Dynamic Characteristics of a Novel Labyrinth Seal with Staggered Helical Teeth Structure. *Alex. Eng. J.* **2021**, *60*, 3177–3187. [CrossRef]
9. Doğu, Y.; Günaydin, I.; Kiliçaslan, Z.; Ileri, T.; Soğanci, S. Design and CFD Analysis of a 150 kW 8-Stage ORC-ROT (Organic Rankine Cycle-Radial Outflow Turbine) and Performance Degradation due to Blade Tip Clearance of Labyrinth Seal. In Proceedings of the ASME Turbo Expo, Oslo, Norway, 11–15 June 2018; Volume 3, p. V003T28A0042018.
10. Dogu, Y.; Sertçakan, M.C.; Gezer, K.; Kocagül, M.; Arican, E.; Ozmusul, M.S. Leakage Degradation of Straight Labyrinth Seal due to Wear of Round Tooth Tip and Acute Trapezoidal Rub-groove. *J. Eng. Gas Turbines Power* **2017**, *139*, 072506. [CrossRef]
11. Watson, C.; Wood, H.G. Developing an Optimal Helix Angle as a Function of Pressure for Helical Groove Seals. In Proceedings of the ASME 2017 Fluids Engineering Division (Publication) FEDSM, Waikoloa, HI, USA, 30 July–3 August 2017; Volume 1A-2017.
12. Stodola, A. *Dampf- und Gasturbinen*; Springer: Berlin/Heidelberg, Germany, 1924.
13. Li, Z.; Li, J.; Feng, Z. Numerical Comparisons of Rotordynamic Characteristics for Three Types of Labyrinth Gas Seals with Inlet Preswirl. *Proc. Inst. Mech. Eng. Part A J. Power Energy* **2016**, *230*, 721–738. [CrossRef]
14. Stachnik, M.; Jakubowski, M. Multiphase Model of Flow and Separation Phases in a Whirlpool: Advanced Simulation and Phenomena Visualization Approach. *J. Food Eng.* **2020**, *274*, 109846. [CrossRef]
15. Bondarenko, G.; Baga, V.; Bashlak, I. Flow Simulation in a Labyrinth Seal. *Appl. Mech. Mater.* **2014**, *630*, 234–239. [CrossRef]
16. Li, Z.; Fang, Z.; Li, J. Numerical Investigation on the Leakage and Rotordynamic Characteristics for Three Types of Annular Gas Seals in Wet Gas Conditions. *J. Eng. Gas Turbines Power* **2019**, *141*, 3. [CrossRef]
17. Pavlenko, I.V.; Simonovsky, V.I.; Pitel', J.; Verbovyi, A.E.; Demianenko, M.M. Investigation of Critical Frequencies of the Centrifugal Compressor Rotor with taking into account Stiffness of Bearings and Seals. *J. Eng. Sci.* **2017**, *4*, C1–C6.
18. Martsynkovskyy, V.A.; Pozovnyi, O.O. The Impact of the Multi-gap Seals on Rotor Dynamics. *J. Eng. Sci.* **2017**, *4*, C7–C12.