Analysis of the geometric parameters influence on the labyrinth seals performance

I Androsovich¹,*, D Borovikov² and M Siluyanova¹

¹Department 12, Moscow Aviation Institute (National Research University), 4 Volokolamskoe Highway, 125993, Moscow, Russian Federation
²Department 2, Moscow Aviation Institute (National Research University), 4 Volokolamskoe Highway, 125993, Moscow, Russian Federation
*E-mail: ivandrosovich@mai.education

Abstract. One of the ways to improve the modern gas turbine engines efficiency is to increase the efficiency of its components. An increase in the gas turbine engine components efficiency may be achieved by reducing air leaks by the seals usage. It is possible to achieve a reduction in air consumption through the seal, including off-design operation conditions, by using modern methods of computational fluid dynamics simulations. One of the possibilities of modern computational fluid dynamics methods is to analyze a simultaneous effect of the large number of variable design parameters. Performance of different labyrinth seals type is analyzed. Direct-flow labyrinth seal is chosen as most common in jet engine design due to its properties. Geometric parameters influences are defined. Optimization resulted in a labyrinth seal configuration that achieves a 20% reduction in seal mass flow. A comparison of the mass flow characteristics of the original and the optimized seal showed that the optimized seal provides reduced mass flow over a wide range of pressure ratios. The results obtained lead to the conclusion about geometric parameters influence on the mass flow and the effectiveness of labyrinth seals optimization to reduce air leaks.

1. Introduction
The work is devoted to the need to improve the turbinomachines of aircraft in order to increase their efficiency by optimizing the characteristics of labyrinth seals to reduce the air leakages [1-3].

The problem of air leakages has been studied by a lot of researchers by now, yet is still relevant for modern jet engines. Various groups of labyrinth seals are considered, depending on the design and location in gas turbine engines [4-6]. The solution to the problem of predicting flow leakage in labyrinth seals was laid by Egli, who proposed a rational solution to the problem of a labyrinth-type device based on the flow parameters characteristic of a pointed object. It may be noted that in one of the modern researches the results of Egli’s model are compared with current computational fluid dynamics results for different tooth shapes and the results are accurate enough for applications that allows hydro-mechanical approach to the problem of flow leakage through labyrinth seals. In the Egli model, the flow assumes an isentropic expansion of the compressible body through an idealized hole (cross-section). Initially, the effects of rotation are neglected, the model includes information about experimentally determined coefficients to account for the transfer of part of the kinetic energy. Empirical values of the transfer coefficient depending on the number of throttles and the gap-to-pitch ratio are used. The flow coefficient is significant for compensation of friction and compression of the flow through the throttles.
Graphs can provide information about changes in the flow rate depending on the number of throttles, gaps, tooth thickness, and overall static pressure indicators.

A mathematical expression for calculating the leakage rate through a labyrinth-type seal is defined [4]. A great contribution to the theory of labyrinth seals was made a review study [5]. It is shown that analytical models for predicting satisfactory performance characteristics of seals are still difficult to regulate. Thus, leakproofness test benches are still used for research purposes.

In recent years, along with the development of computational fluid dynamics methods, many computational studies have been performed and models have been developed for analyzing flows inside labyrinth seals [6-10]. In research paper [10], scientists came to the development of new configurations of labyrinth-type seals. Their characteristics were investigated [11] and the results obtained show that a good compaction configuration should be able to provide a high rate of kinetic energy dissipation.

The newest research on labyrinth seals geometry influence and seal geometry optimization are made in [12], it provides a lot of data on labyrinth seals geometry parameters influence including gap size and comb length. An optimization of seal with CFD methods is studied in [13]. Thus, this works are close to the current research and to previous part of it [14].

Based on a review of the literature and experience, it is clear that efforts to improve the performance of labyrinth compaction should be directed either to the development of a new configuration or to the improvement of existing structures. During current research an application of labyrinth seals for gas turbine engines is considered. Main applications are defined as turbomachines and rotors of aircraft engines. The generalized configuration of the labyrinth seal is chosen.

In the course of a series of calculations, the influence of the mesh elements size and the configuration of the prismatic sublayer on the calculation results is evaluated. Various continuous and discrete geometric parameters influence is defined and seal optimization is performed.

2. Calculations

To perform an optimization and the response analysis Ansys CFX© is used.

2.1. Mathematical model

The calculation of labyrinth seals is based on modeling equations based on the classical method of Navier-Stokes equations and the following expressions that represents fundamental conservation lows conservation of mass (1), conservation of momentum (2) and conservation of energy (3), equations (4) and (5) describes the turbulence model and thermodynamic equations of state (6) end enthalpy (7).

Continuity equation:

\[ \frac{\partial \rho}{\partial t} + \nabla (\rho U) = 0 \]  

where \( \rho \) – density; \( t \) – time; \( U = (U_x; U_y; U_z) \) – velocity vector.

Momentum equation (2):

\[ \frac{\partial \rho U}{\partial t} + \nabla (\rho U \otimes U) = -\nabla p + \nabla \tau + S_m \]  

where \( \tau \) – stress tensor; \( p \) – pressure; \( \otimes \) – tensor product; \( S_m \) – momentum source.

Energy equation:

\[ \frac{\partial \rho H}{\partial t} - \frac{\partial \rho H}{\partial x_i} + \nabla (\rho U H) = \nabla \left( \lambda \nabla T \right) + \nabla (U \tau) + S_E \]  

where \( H \) – total enthalpy; \( T \) – temperature, \( k \) – turbulent kinetic energy; \( \lambda \) – thermal conductance coefficient; \( S_E \) – source component.

SST turbulence model general equations:

\[ \frac{\partial P_k}{\partial t} + \frac{\partial}{\partial x_j} \left( \rho U_j k \right) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma} \right) \frac{\partial k}{\partial x_j} \right] + P_k - \beta' \rho \omega + P_{kb} \]
where \( \sigma, \alpha \) and \( \beta \) are constants; \( \mu \) – kinematic viscosity.

Equation of state of an ideal gas:

\[
p = \rho RT = \frac{RT}{\varrho}
\]

where \( R \) – gas constant.

The equation of enthalpy:

\[
h - h_{\text{ref}} = \int_{T_{\text{ref}}}^{T} C_p(T) dT
\]

where \( h \) – enthalpy; \( h_{\text{ref}} \) – enthalpy of formation; \( C_p \) – isobaric heat capacity.

2.2. Boundary conditions

The open boundary condition with specified pressures was used as the boundary conditions (figure 1). The performance of two boundary conditions for modeling a sector are analyzed. A rotational periodicity condition did not work for certain geometry configurations, which is probably due to the low height of the channel relative to its radius. An alternative was to use the free slip wall condition. This condition provided much greater robustness of the calculation series, but is less accurate due to prevention of circumferential movement of gas in the seal.

Rotational periodicity boundary condition appeared 3% errors (out of 200 calculations), free slip wall boundary condition 0.1% errors (out of 300 calculations) with the difference in results lower than 2%.

Analysis is performed with a pressure ratio in a range from 1.1 to 1.7 and low-pressure side pressure 1 atm. During current research no rotation was applied.

2.3. Optimization problem definition

During the optimization an optimal geometrical parameters combination is looked for. Mass flow through the labyrinth seal is chosen as an optimum criterion. The problem of the labyrinth seal optimization may be formulated in terms of set theory by set of seals (8) and set of optimal seals (9) as follows:

\[
\mathbb{S} = \left\{ S_i \mid P_{il} \leq P_i \leq P_{ih} \right\}
\]

\[
\bar{S} \in \mathbb{S} : \min \left\{ f\left(\bar{S}_i\right) \right\} = f\left(\bar{S}_i\right)
\]

Figure 1. Boundary conditions rotational periodicity/free slip wall (red) and opening pressure (blue).

Figure 2. Labyrinth seal configuration.
where $S$ is a set of seal geometrical parameters; $S$ – seal configuration; $P$ – geometrical parameter; $f (S)$ – an optimality criterion.

2.4. Geometry and mesh

2.4.1. The seal configuration selection. The generalized configuration of the labyrinth seal is selected. Several continuous geometric parameters of the seal are taken [13]. Continuous geometric parameters of the seal (figure 2): two sides angle of inclination, fillets radius at the base, the height of the ridge, the width of the comb, and width of the top of the ridge. Additional seal geometric parameters: seal gap (default gap 0.5 mm) and the number of ridges (a discrete parameter). List of seal parameters and their values is presented in table 1.

Table 1. Geometry configuration.

| Variable parameter name                        | Default value | Minimum value | Maximum value |
|------------------------------------------------|---------------|---------------|---------------|
| Top face length, mm                           | 0.5           | 0.1           | 0.6           |
| Fillets radius, mm                            | 0.6           | 0.1           | 0.6           |
| Ridge high, mm                                | 2             | 2             | 3             |
| Gap size, mm                                  | 0.5           | 0.25          | 0.6           |
| High pressure side angle of inclination, deg  | 105           | 90            | 120           |
| Low pressure side angle of inclination, deg   | 90            | 90            | 120           |
| Distance between the ridges, mm               | 0.42          | 0.4           | 0.8           |
| Number of ridges                              | 3             | 3             | 5             |

To save computing resources, the compaction sector of 2 degrees is calculated. The results are compared with the sector of 10 and 30 degrees and showed no difference.

2.4.2. The mesh convergence analysis. A multizone mesh with a hexadominant type and a prismatic sublayer is chosen. In the course of a series of calculations, the influence of the size of the mesh elements and the configuration of the prismatic sublayer on the calculation results was evaluated, the results are shown in table 2. A mesh convergence analysis was performed for the initial compaction.

The result of the analysis showed that a grid with an element size of less than 60 microns should be used, which provided at least 4 elements in the channel in addition to the prismatic sublayer. The analysis also showed a significant effect of using the wall function on the calculation result, which forced the use of a prismatic sublayer with a height of the first element of 1 µm and 20 layers that allows to have growth rate up to the size of the main elements less than 1.25 (figure 3).

Table 2. The mesh convergence analysis results.

| Average element size, mm | Number of elements | Relative number of elements | Air consumption, g/s | Relative air consumption | Yplus max |
|--------------------------|--------------------|---------------------------|----------------------|-------------------------|-----------|
| 0.1                      | 88230              | 0.165908                  | 9.81                 | 1.008222                | <50       |
| 0.06                     | 205050             | 0.385577                  | 9.89                 | 1.016444                | <50       |
| 0.05                     | 280470             | 0.527398                  | 9.91                 | 1.018499                | <50       |
| 0.04                     | 417920             | 0.785859                  | 9.97                 | 1.024666                | <50       |
| 0.07                     | 221600             | 0.416698                  | 9.58                 | 0.984584                | <1        |
| 0.06                     | 283680             | 0.533434                  | 9.63                 | 0.989723                | <1        |
| 0.05                     | 370320             | 0.696352                  | 9.59                 | 0.985612                | <1        |
| 0.04                     | 531800             | 1                         | 9.73                 | 1                       | <1        |
3. Results and discussion

3.1. Geometric parameters influence

To analyze the geometrical parameters influence, second order response surface is calculated by central composite experimental design. The results of response for different parameters on the central part of multidimensional response surface are shown on the following charts with the influence analysis.

Sharp top edge is more effective to reduce mass flow thru the seal due to radial velocity component appearing on a ridge side faces that breaks jet stream core on a ridge top. Wide top face could reduce a mass flow as well due to boundary layer accumulation that reduces the effective cross section area in the seal gap, but increased width of top face also leads to increased length and weight of the seal. Level of top face width influence on a mass flow is about 2.5% (figure 4a). Fillets radius at the base of the ridge does not significantly affect the mass flow and could be selected by technological reasons (figure 4b).

![Figure 3. Computational mesh](image)

![Figure 4. Mass flow as a function of the length of the ridge top face (a) and the fillet radius at the base of the ridge (b).](image)

Higher ridges are more effective in seal mass flow reduction. Higher ridges are leaving more space for an eddy structures, that leads to increased eddies intensity (figure 5a). The decrease of the gap substantially reduces the flow of air due to effective cross area reduction (figure 5b). Achieved results for a gap size influence showed a good coincidence with reference data [12] (figure 6).
Figure 5. Mass flow as a function of the ridge height and the gap size value.

Figure 6. Influence of the relative mass flow in comparison with reference data.

Higher side faces angle of inclination allows to reduce mass flow thru the seal. On a high pressure side, the reduction is increasing up to 120 degrees due to increasing eddy structure intensity and appearing of radial velocity component on a leading edge of the gap (figure 7a). On a low pressure side, the reduction continues with over 120 degrees face angles (figure 7b). Low pressure side inclination increases the eddy structures intensity. The disadvantages of this parameters are a high influence of a seal size and weight.

Figure 7. Mass flow as a function of angle of inclination of the ridge on the high pressure and low-pressure sides.

Increase of a comb length also leads to increasing in eddy structures intensity and allows to reduce mass flow (figure 8).
Increase the number of ridges significantly reduces mass flow thru the seal (figure 9).

Simulation results allows to conclude that all analyzed parameters have two mechanisms of seal characteristics improvement. The first one is an energy dissipation, and the second is a reduction of the effective cross section area. Ridge high, slope faces and comb length leaded to the increased eddies intensity between the ridges and as a result to the dissipation of energy and reduction of flow velocity in the gap. Additional ridges increase a total energy dissipation in the seal. Gap size affect the cross-section area directly and top face length use both mechanisms depending on its length. Fillet radius does not affect the flow I analyzed seal. Continues parameters sensitivity is presented (figure 10)

3.2. Labyrinth seal optimization performance comparison

Obtained result allowed to perform a seal optimization. During optimization a mass flow is chosen as an optimality criterion.

Comparison of the original (figures 10-11) and optimized (figures 12-14) seal flows showed that optimized seal has a jet stream breaking on a top of each ridge and highly growed up eddy structures intensity between each pair of ridges, this led to the increased energy dissipation, significant reduction in gap flow velocity and as a result to reduction of mass flow via seal up to 28%.

The increased distance between the ridges tops provides more space for eddy structures and allows to increase their intensity due to the action of viscous forces between the flow core and the gas in the comb; the angle of inclination of the ridge surface and, to a lesser extent, the distance between the ridges, had a key role in increasing the distance between the ridges top faces. Increased height of the ridge also provided more space to form a more intense eddy structure. The sharp top and a certain angle of inclination of the ridge on the high-pressure side contributed to appearing of a significant radial velocity component in the gap, which also contributed to the deceleration of the flow.
3.3. Seals performance comparison

Further comparison of initial and optimized seals with various pressure ratios is performed to analyze the reduction of mass flow in the optimized seal under a wide range of operating conditions (table 3, figure 15).
Figure 15. Relative mass flow of the original (blue) and optimized (orange) seal depending on a pressure ratio.

Table 3. Flow characteristics of the original and optimized seal.

| Pressure ratio | Relative air consumption of original seal | Relative air consumption of the optimized seal |
|---------------|------------------------------------------|---------------------------------------------|
| 1.1           | 1                                        | 0.723                                       |
| 1.15          | 1.217                                    | 0.903                                       |
| 1.2           | 1.417                                    | 1.024                                       |
| 1.3           | 1.755                                    | 1.319                                       |
| 1.4           | 2.041                                    | 1.544                                       |
| 1.5           | 2.293                                    | 1.739                                       |
| 1.6           | 2.521                                    | 1.941                                       |
| 1.7           | 2.732                                    | 2.139                                       |

A comparison of seals under different operating conditions showed that the high quality of the optimized seal is maintained in a wide range of operating conditions and provides a reduction in consumption of more than 20% compared to the initial configuration.

4. Conclusions

Several conclusions are following from the performed research:

The method of optimization of labyrinth compaction is presented. Different geometrical parameters of the labyrinth seal have a various influence on its performance, an influence of several parameters is estimated.

The mesh convergence and mesh influence analysis showed that when simulating the operation of the labyrinth seal, it is necessary to use mesh that does not allows the usage of logarithmic wall function and provides the values of $y+ \approx 1$ as well as has enough elements in the gap. Optimization of the labyrinth seal allows to significantly reduce the flow rate through it. A comparison of the flow characteristics of the original and optimized seals showed that the optimal seal provides less mass flow in a wide range of pressure ratios. The highest quality direct flow labyrinth seal reduces the gap loss by up to 50% depending on the gap size.

References

[1] Sun Y, Kolesnik S A and Kuznetsova E L 2020 Mathematical modeling of coupled heat transfer on cooled gas turbine blades. *INCAS Bulletin*. 12 193 https://doi.org/10.13111/2066-8201.2020.12.S.18

[2] Vasiliev V S, Levochkin P S, Chvanov V K and Timushev 2019 Proposals for improving the
efficiency and durability of the turbines of turbo-pump assemblies in liquid-propellant rocket engines by using double-sided crest-type radial labyrinth seals. *IOP Conf. Ser.: Mater. Sci. Eng.* **491** 012018 https://doi.org/10.1088/1757-899X/491/1/012018

[3] Aleksandrovskaia L N, Kerber O B, Iosifov P A, Pankina G V and Boitsov B V Using mixture distributions for the analysis and estimation of ultra-low risks in the problems of ensuring the safety of automatic aircraft landing. *Russian Aeronautics** **62**(2) 199 https://doi.org/10.3103/S1068799819020041

[4] Tong S K and Kyu S C 2009 Comparative analysis of the influence of labyrinth seal configuration on leakage behaviour. *J. Mech. Sci. Technol.* **23** 2830 https://doi.org/10.1007/s12206-009-0733-5

[5] Morrison G and Chi D 1985 Incompressible flow in stepped labyrinth seals. *ASME/ACSE Applied Mechanics, Bioengineering and Fluids Engineering Conf.* (June 24-26, Albuquerque, New York: ASME) p 8

[6] Schram V, Willenborg K, Kim S and Wittig S 2002 Influence of a honeycomb facing on the flow through a stepped labyrinth seal. *J. Eng. Gas Turb. Power* **124** 140 ASME-Paper 2000-GT-291

[7] Bidkar R, Edip S, Jifeng W, Azam T, Andrew M, Maxwell P, Grant M, Timothy A and Jeffrey M. 2016 Low-leakage shaft-end seals for utility-scale supercritical CO₂ turboexpanders. *J. Eng. Gas Turb. Power* **139** 022503 https://doi.org/10.1115/1.4034258

[8] Soemarwoto B, Kok J C, Cock K M J, Kloosterman A B and Kool G A 2007 Performance evaluation of gas turbine labyrinth seals using computational fluid dynamics. *Proc. GT2007 ASME Turbo Expo 2007: Power for Land, Sea and Air* (14-17 May, Montreal, Canada: National Aerospace Laboratory NLR) p 1553

[9] Wang W, Liu, Y, Jiang P and Chen H 2007 Numerical analysis of leakage flow through two labyrinth seals. *J. Hydrodyn.* **19**(1) 107 https://doi.org/10.1016/s1001-6058(07)60035-3

[10] Vakili A D, Meganathan A J, Michaud M A and Radhakrishnan S 2005 An experimental and numerical study of labyrinth seal flow. *Proc. GT2005-68224 ASME Turbo Expo 2005: Power for Land, Sea and Air* (June 6-9, Reno-Tahoe, Nevada, USA: ASMEDC) p 1347

[11] Vakili A D, Meganathan A J, Ayyalasomayajula S and Stephen H 2006 Advanced labyrinth seals for steam turbine generators. *Proc. Of GT2006 ASME Turbo Expo 2006: Power for Land, Sea and Air* (May 8-11, Barcelona, Spain: ASME) p 1599

[12] Zhigang L, Jun L and Zhenping F 2016 Labyrinth seal rotordynamic characteristics part i: geometrical parameter effects. *Journal of Propulsal and Power* **32**(5) 1 https://doi.org/10.2514/1.B35817

[13] Tyacke J C, Dai Y, Watson R and Tucker P G 2021 Design optimisation of labyrinth seals using LES. *Math. Model. Nat. Pheno.* **16** 1 https://doi.org/10.1051/mmm/2020056

[14] Androsovich I V and Siluyanova M V 2021 Optimization of labyrinth seals in gas-turbine engines. *Russian Engineering Research* **41**(4) 360 https://doi.org/10.3103/S1068798X21040043