ABSTRACT

The major sealing device between rotating and static aeroengine parts is still of labyrinth type. Nowadays the fins and the stator may be coated and the running conditions very often cause heavy rubs which leads to severe surface imperfections.

This paper investigates the influence of rounded fins and worn coatings on the discharge coefficients of straight through labyrinth seals. The relevant effects of Reynolds number are investigated.

Some measured fin and coating surfaces from engine parts are presented. Experimental and numerical results show a strong effect of wear on labyrinth seal performance.

1 INTRODUCTION

Among the different types of sealing devices in aeroengines, the labyrinth seals are still dominant especially for large radii with high circumferential speeds.

The following requirements have to be met:

a) low leakage rate
b) high reliability
c) low heat generation
d) applicability at high temperatures and circumferential speeds

The advantage of labyrinth seals lies positively in points b) to d).

Regarding the leakage rate they cannot compete with most of the other sealing devices, but only a lower fraction of the cooling air is controlled by seals (blade and vane cooling not) and some of the leakages may be used in the downstream system. The point "low heat generation" mainly applies to bearing chamber seals and is especially important for turbine bearing chambers with the hotter environment. The cooling effectiveness has been improved in the past, but the amount of cooling air cannot be reduced below a level, where several functional requirements are to be warranted under all operating conditions. Taking all that into account, labyrinth seals will well be competitive in future.

Labyrinth seals never run exactly to nominal conditions for which all the experimental and numerical investigations are done, but the literature mainly is confined to the ideal conditions, in Trutnovsky and Komotori, 1981, the influence of rounded fins is highlighted but not for different Re-numbers.
There is now quite an amount of papers which documents the use of CFD for investigation of labyrinth seal behaviour. Rhode and Sobolik, 1986; Rhode and Hibbs, 1989 and 1992; Rhode and Nail, 1992 and Rhode and Guidry, 1993. Furthermore Wittig et al, 1983; Wittig et al, 1987, compared CFD calculations with model experimental results including heat transfer. Sturgess, 1988, highlighted several aspects of the application of CFD to labyrinth seals. Brownell et al, 1988, applied non-intrusive methods for investigating labyrinth seal flow, the holographic pictures were used to obtain a better insight into the complex flow mechanism. Morrison et al, 1992, presented experimental data for the verification of CFD calculations for a secondary recirculation zone in a labyrinth seal. The standard labyrinth test data from the classical literature were compared among each other and with a new method by Zimmermann and Wolff, 1987.

2 METHODS

2.1 FLOW FIELD COMPUTATION

The flow field within the labyrinth seals was computed with a commercially available CFD code. The numerical method of the "Task Flow" code is described in Raw et al, 1989. Relevant to this investigation is:

- turbulent eddy viscosity with the standard k-ε model

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>geometrical cross sectional area</td>
<td>m²</td>
</tr>
<tr>
<td>C</td>
<td>discharge coefficient</td>
<td>-</td>
</tr>
<tr>
<td>K</td>
<td>surface roughness</td>
<td>mm</td>
</tr>
<tr>
<td>m</td>
<td>mass flow rate</td>
<td>kg/s</td>
</tr>
<tr>
<td>n</td>
<td>number of fins</td>
<td>-</td>
</tr>
<tr>
<td>Pₛ</td>
<td>static pressure</td>
<td>N/m²</td>
</tr>
<tr>
<td>Pₜ</td>
<td>total pressure</td>
<td>N/m²</td>
</tr>
<tr>
<td>Pₜmax</td>
<td>maximum local total pressure in velocity profile</td>
<td>N/m²</td>
</tr>
<tr>
<td>Q</td>
<td>mass flow function</td>
<td>kg/s</td>
</tr>
<tr>
<td>r</td>
<td>rounding radius</td>
<td>mm</td>
</tr>
<tr>
<td>R</td>
<td>gas constant</td>
<td>J/kg K</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
<td>-</td>
</tr>
<tr>
<td>s</td>
<td>seal clearance</td>
<td>mm</td>
</tr>
<tr>
<td>T</td>
<td>total temperature</td>
<td>K</td>
</tr>
<tr>
<td>l</td>
<td>whole labyrinth</td>
<td></td>
</tr>
<tr>
<td>d</td>
<td>downstream</td>
<td></td>
</tr>
<tr>
<td>k</td>
<td>rounded plus surface roughness</td>
<td></td>
</tr>
<tr>
<td>r</td>
<td>rounded</td>
<td></td>
</tr>
<tr>
<td>s</td>
<td>sharp edged</td>
<td></td>
</tr>
<tr>
<td>u</td>
<td>upstream</td>
<td></td>
</tr>
</tbody>
</table>

Subscripts:

id, id | ideal
b, | downstream
k, | rounded plus surface roughness
l, | whole labyrinth
r, | rounded
s, | sharp edged
u, | upstream

Fig. 1: Geometry of a 3-fin straight through labyrinth seal
a) overall dimensions (t = 2.4 mm; b = 0.25 mm; h = 2.1 mm)
b) section of BFC-grid used for rounded fin geometry (BFC = 'body fitted coordinates')
- a high degree of numerical robustness is guaranteed by use of a fully coupled linear solver, accelerated by an additive correction multigrid scheme and a block correction scheme
- accuracy is high due to a fully implicit, co-located finite volume method with a flux element-based discretization of geometry and the availability of a second-order discretization

The plain computational grid models an existing labyrinth seal with a scale of 1:1. An upstream and downstream addendum is necessary because of convergence aspects.

The single-block grid, a segment of which is shown on Fig. 1, is of H-type. Walls are closely described by boundary fitted coordinates. Near surface grid refinement is used. A flexible grid block structure was developed to cover the different geometries with 360 x 100 nodes.

2.2 LABYRINTH FLOW CORRELATIONS

Some of the geometrical parameters are shown in Fig. 1. Labyrinth flow characteristics are mostly described by adding loss coefficients to the gas-dynamic throughflow correlations for restrictors in series.

\[
Q = \frac{m_1 \sqrt{T}}{C_{1, \text{geom}}} = f \left( \frac{P_{t,u}}{P_{s,d}}, n \right) \quad (1)
\]

for the labyrinth as a whole \((Q_l)\) or for the individual fin.

For an ideal labyrinth the following linearized correlation is used:

\[
Q_{1, \text{id}} = \frac{m_{1, \text{id}} \sqrt{T}}{A_{\text{geom}}} = \sqrt{\frac{1 - (P_{s,d}/P_{t,u})^2}{R(n+\ln(P_{t,u}/P_{s,d}))}} \quad (2)
\]

From the measured or numerically determined \(m_1\) and \(m_{1, \text{id}}\) from equ. (2) the loss coefficient \(C_{1}\) is determined:

\[
C_{1} = \frac{m_1}{m_{1, \text{id}}} \quad (3)
\]

where \(C_{1}\) could be subdivided into the contraction and the carry over coefficient, but this has not been applied in this paper. For straight through seals \(C_{1}\) may be above one, because of the carry over effect.

2.3 METHOD OF EVALUATION

To get a deeper understanding of the complex flow phenomena and loss mechanism a combined method of investigation and evaluation has been adopted. Experimental and numerical studies are compared with respect to their overall characteristics; CFD post-processing is used to trace local effects; individual discharge coefficients evaluated for each fin allow to draw conclusions for the distribution of pressure losses. The overall throughflow functions equation (1) to (3) are used to equate the labyrinth seal mass flow, which is the only quantity of interest for most of the practical applications.

The individual coefficients of the fins are used to crosscheck the overall coefficients and to find the reasons for the trends. Together with vector plots from CFD a better understanding of the flow physics is achieved. It should be noted that there is a slight difference to the method used in Zimmermann and Wolff, 1987, there the individual \(C\)-values were based on a plane at fin exit. In this paper rounded fins are as well incorporated therefore a reference plane in the middle of the fin had to be chosen.

It cannot be hoped to obtain generalized methods for worn seals, as the type of deviation from the ideal labyrinth geometry will be manifold. In this paper numerical and experimental results are presented for rounded fins and rough stator surfaces with and without grooves, assuming the same geometrical condition for every fin.

Fig. 2: Labyrinth fin cross sections derived from an engine serial part: a) typically new labyrinth; b) after some running time in an engine
For arbitrary geometry distortions a method has to be adopted which evaluates every fin on its own, similar to the theory explained in Zimmermann and Wolff, 1987, where the first fin was individually treated. But the application of such a method would be limited as normally the sizes of the distorted geometry is not known.

3 EFFECT OF FIN RADIUS ON LABYRINTH FLOW

Labyrinth seals for aeroengines are normally designed sharp edged with small corner radii, the minimum of which are dictated by the manufacturing process and nowadays as well by design to cost rules. But it is anyway of no use to aim for too small radii as seal deterioration during engine operation starts after a few running hours. Due to relative movements of the rotor against the stator during transient conditions, running in of the seal fins into the coating can often not be avoided, since otherwise big seal clearances would result.

Fig. 2 shows a comparison of the fin cross section as per design scheme with a worn condition, measured after engine strip. Fig. 3 shows a fin without coating and with the coating on, before and after engine testing. The coated fin is in both cases fully rounded.

A series of investigations on labyrinth seals which are designed to run into a coating, revealed, that the fins are fully rounded after a short running time. Therefore the relevant effect on throughflow should be known. Model tests described in the literature were all done with nearly sharp edged fin corners, but in most cases small radii, which can not be avoided, were not reported and very likely not even measured.

3.1 EFFECT OF CORNER RADIUSING ON ORIFICE FLOW

Orifices show a large flow increase with increasing inlet corner radius, see Fig. 4, curve a. Numerous papers have been published dealing with this subject. The results from Idel'chick, 1966, have been identified as a mean basis, they lie well in the middle of the scatter band taking test results of many authors. Furthermore most of the interesting range of dimensionless corner radii is covered (for an orifice the hydraulic diameter d is used instead of 2s).

The maximum possible discharge coefficient would be nearly one and the minimum is approximately $C = 0.6$ for a sharp edged orifice, giving a margin for corner radiusing of approximately 65%.

3.2 EFFECT OF CORNER RADIUSING ON LABYRINTH FLOW

Labyrinth seals basically consist of a series of annular constrictions. They are characterized by several loss mechanisms. The pressure losses can be mainly divided into friction losses at the stator wall, flow separation (if existing) at the fin tip and subsequent diffusion and losses arising from the big vortex in the expansion chamber.

The influence of various inlet and outlet duct configurations is not considered in this paper. Cor-
ner radius effects in labyrinths were expected to be similar to those for orifices and in fact because of lack of better data, for a long time, orifice corner rounding data were used for labyrinths, although the basic flow pattern is different. Whereas orifices show a symmetrical flow separation and little effect of wall friction the labyrinth flow shows a single side flow separation combined with friction losses at the static wall.

### Discharge Coefficients

<table>
<thead>
<tr>
<th>s (mm)</th>
<th>Re-no.</th>
<th>r (mm)</th>
<th>1st fin</th>
<th>2nd fin</th>
<th>3rd fin</th>
<th>labyrinth</th>
<th>1/C12**2</th>
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<tr>
<td>0.2</td>
<td>476</td>
<td>0.0</td>
<td>0.68</td>
<td>0.70</td>
<td>0.70</td>
<td>1.03</td>
<td>0.91</td>
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<td>0.2</td>
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<td>0.68</td>
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<td>0.70</td>
<td>1.10</td>
<td>0.83</td>
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<td>0.80</td>
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<td>0.0</td>
<td>0.75</td>
<td>0.71</td>
<td>0.71</td>
<td>1.10</td>
<td>0.83</td>
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<td>1.12</td>
<td>0.80</td>
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<td>0.81</td>
<td>0.84</td>
<td>1.16</td>
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<td>0.2</td>
<td>19000</td>
<td>0.10</td>
<td>0.80</td>
<td>0.81</td>
<td>0.83</td>
<td>1.14</td>
<td>0.77</td>
</tr>
<tr>
<td>0.4</td>
<td>716</td>
<td>0.10</td>
<td>0.71</td>
<td>0.70</td>
<td>0.69</td>
<td>1.12</td>
<td>0.80</td>
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<td>0.73</td>
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<td>0.77</td>
<td>0.78</td>
<td>1.21</td>
<td>0.68</td>
</tr>
</tbody>
</table>

Tab. 1: Effect of corner radius and stator grooves on the discharge coefficient of a smooth 3-fin straight through labyrinth

### 3.2.1 EXPERIMENTAL RESULTS

The effects of fin corner radiusing for a 3-fin straight through labyrinth seal with the geometry of Fig. 1 are shown in Fig. 4, curve c. These test data (not yet published) were obtained by Wittig, using the test rig described by Wittig et al, 1982. Further data in Fig. 4 curve b are taken from company owned data for a 3-fin stepped labyrinth seal. The straight through seal shows radius effects, which are much lower than those for orifices and also lower than those for stepped seals. The stepped seal behaves more like orifices in series with a full "dynamic head" loss behind each fin, as the carry over effect is mostly negligible.

### 3.2.2 RESULTS OF NUMERICAL CALCULATIONS

In Tab. 1 the computed discharge coefficients as well as a pressure loss coefficient resulting from the overall labyrinth pressure loss are listed. The latter coefficient 1/C12 is plotted against the Reynolds number in Fig. 5 and 6.

1/C12 has been chosen as a pressure loss term, it is equivalent to a friction coefficient.

![Fig.5: Effect of corner radius on the discharge coefficient of a 3-fin labyrinth for s = 0.2 mm](image)

![Fig.6: Effect of seal clearance and stator grooves on the loss coefficient of a 3-fin straight through labyrinth with rounded fin corners; r = 0.1 mm](image)

At low Re-numbers the wall friction determines the pressure loss and straight lines, very similar to the Blasius law for laminar pipe flow, could be seen in a log/log-scale.

The vector plot in Fig. 7 for sharp edged fins and laminar flow shows that the flow contraction above each fin is small. Here the first fin plays not a dominant role as described by Zimmermann and Wolff, 1987, for turbulent flow. Fig. 8 shows vector plots
for turbulent flow over sharp edged fins. Above the first fin there is a significant flow contraction, whereas above the other two fins not. This is because of the different approach flow conditions. From Tab. 1 it can be seen, that the discharge coefficient for the first fin, in case of laminar flow, may even exceed those for the other two fins. The stator wall boundary layer thickness, which is not yet grown to the full gap size above the second fin, explains this phenomenon.

As shown in Fig. 6 and Tab. 1, the discharge coefficients increase with the seal clearance (compare $s = 0.4 \text{ mm}$ with $s = 0.2 \text{ mm}$). This is as expected and confirms the influence of the "carry over" effect as it is modelled in all of the numerous correlations, which are in daily use.

It is not understood, why in the turbulent case there is a flat minimum (Fig. 5) at approximately $Re = 10,000$, but this was observed as well experimentally, see Trunovsky and Komotori, 1981. From Tab. 1 it can be seen, that the effect is mainly caused by the first fin.

For turbulent flow and rounded seal fins a few test points are plotted in Fig. 5 and 6, marked with dark symbols. Compared to the numerical results the level of the pressure loss coefficients agrees quite well.

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4 Effect of Surface Roughness and Grooves on Labyrinth Flows

In aeroengines the stator wall of a labyrinth seal is mostly fitted with special coatings. These coatings have to fulfill different functions, as there are:
- to enable fin penetration without fin wear
- to avoid erosion and corrosion
- to be temperature resistant even at fin contact

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Fig. 7: Vector plot for a 3-fin straight through labyrinth seal ($Re = 560; s = 0.20 \text{ mm}; r = 0.0 \text{ mm}$)

Fig. 8: Vector plot for a 3-fin straight through labyrinth seal ($Re = 16100; s = 0.20 \text{ mm}; r = 0.0 \text{ mm}$)
to show good reliability and maintainability

Due to the accepted fin running in and to the not fully avoidable erosion and corrosion processes, after a certain engine running time the coating surface shows grooves, break-outs and extreme roughness.

![Fig. 9: Typical worn labyrinth seal stator coating (top and bottom cross sections)](image)

Since surface roughness and even grooves are real effects, the assumption of nominal conditions for labyrinth flow calculation, i.e. smooth stator surface, no longer remains fully correct. An example of an engine surface measurement is presented in Fig. 9 for a sprayed NiC coating at two circumferential positions.

The engine running time to develop such coating surfaces strongly depends on the position of the labyrinth seal inside the engine. The main parameters of influence are radial excursions of the rotor relative to the stator, air flow with a high preswirl and particle contamination of the air flow.

4.1 EFFECT OF GROOVES

The typical appearance of labyrinth seal wear in an aero engine combines several geometrical deviations. All such differences from nominal conditions

![Fig. 10: Vector plot for a 3-fin straight through labyrinth seal (Re = 19000; s = 0.20 mm; r = 0.10 mm)](image)

![Fig. 11: Vector plot for a 3-fin straight through labyrinth seal with stator grooves (Re = 4750; s = 0.20 mm; r = 0.10 mm)](image)
lead to changed sealing characteristics for the labyrinth. In Tab. 1 the results of numerical calculations are summarized showing the effect of grooves in the stator wall. The investigation was done with rounded seal fins (\( r = 0.1 \, \text{mm} \)). The seal clearance relative to the stator wall was kept constant at \( s = 0.2 \, \text{mm} \). The grooves were simulated by rectangular shapes with a groove width of 0.75 mm and a groove depth of 0.2 mm. Fig. 6 shows the effect of such a groove size on the pressure loss coefficient for a 3-fin straight through labyrinth with well rounded fins.

The velocity plots in Fig. 10 and 11 present an overview of the flow distribution across a seal fin with and without a groove in the stator.

The results indicate, that the distance of the groove side faces to the fins restrict the flow, hence producing a lower pressure loss, when compared with the results for \( s = 0.4 \, \text{mm} \) without grooves (possibly by not putting the coating on). As this depends very much on the dimensions of the grooves, it will be very difficult to provide generalized correlations for these cases.

### 4.2 EFFECT OF SURFACE ROUGHNESS

The influence of the surface quality on the friction losses for pipe flow is well known and presented for example in the Moody-diagram. To illustrate the influence of the surface roughness on the labyrinth characteristic, numerical calculations for a relative surface roughness of \( K/2s = 0.25 \) were performed. They have been scaled from an unpublished investigation with the logarithmic law of the wall. The results are compared with "smooth stator wall" calculations in Fig. 5. The obtained trend is similar to pipe flow characteristics, as would be expected for the turbulent flow regime. For instance there is a flat minimum for a large wall roughness at a Re-number of \( 10^4 \).

### 5 ENGINE ASPECTS

The secondary air system of an aeroengine highly relies on the characteristic of the sealing devices. The engine performance and the integrity of engine parts and the engine itself depends on the proper functioning of this air system.

The range of Reynolds number for labyrinth seal throughflow in a typical fighter engine can be seen in Tab. 2. The flight envelope of such an engine includes high altitude flight conditions corresponding to very low Re-numbers, as well as low level high speed mission points with high Re-numbers.

The labyrinth seal in an aero engine has to be calculated carefully for both flow regimes to avoid any adverse effect at any flight condition.

In the design phase of an engine, seal deterioration will be taken into account and the secondary air system design can be adjusted accordingly to minimize its influence.

<table>
<thead>
<tr>
<th>mission point</th>
<th>bearing chamber seal</th>
<th>turbine interstage seal</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>straight through type</td>
<td>stepped type</td>
</tr>
<tr>
<td>high level low speed</td>
<td>100</td>
<td>1000</td>
</tr>
<tr>
<td>SL take off</td>
<td>2000</td>
<td>10000</td>
</tr>
<tr>
<td>low level high speed</td>
<td>5000</td>
<td>20000</td>
</tr>
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</table>

Tab. 2: Typical range of Reynolds-numbers for labyrinth seals in a modern fighter engine.

### 6 CONCLUSIONS

- In the literature more or less sharp edged labyrinth fins are used for all investigations up to now.
- The real existing small rounding radii in case of experimental investigation have not been reported in most publications.
- For turbulent flow, the influence of a small rounding has already a significant effect.
- For worn engine conditions the fins are fully rounded, their discharge coefficients can be 35% above a sharp edged and approximately 15% above a newly manufactured stepped labyrinth seal. For a straight through labyrinth seal the figures are 15 and 6%.
- Even in the laminar flow regime (high altitude) the influence of surface imperfections and rounding of fins cannot be neglected.

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8
Roughness of the surface reduces the leakage flow by 8% for turbulent flow and a roughness, which is equivalent to measured case (Fig. 9).

Grooves in the surface can increase the labyrinth leakage by considerable amount. But the leakage will be in most cases less than without a coating and an equivalent higher gap size.

At high altitude flight conditions the labyrinth leakages are reduced. For instance in the extreme case of Re-numbers between 5000 and 100 the friction loss is increased by approximately 20%.

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