WINDAGE HEATING OF AIR PASSING THROUGH LABYRINTH SEALS

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ABSTRACT

The viscous drag on rotating components in gas turbine engines represents both a direct loss of power from the cycle and an input of heat into the secondary (cooling) air system. Hotter cooling air in turn means increased flow requirements. The effects of windage on performance are therefore compounded.

To facilitate accurate temperature predictions of highly stressed components, information is needed on windage characteristics of all elements in the secondary cooling system. Much information is available in the literature for discs, cones, cylinders, bolts etc but little has been published on windage heating in high speed seals.

Results are presented for experiments carried out (at representative non-dimensional conditions) on different designs of labyrinth seals. The results are compared with values calculated from the simple momentum balance theory suggested by McGreeham and Ko (1989) and with several values determined from CFD analysis.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Variable</th>
<th>Units</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>m²</td>
<td>Leakage area</td>
</tr>
<tr>
<td>c</td>
<td>m</td>
<td>Seal radial clearance (r s - r R)</td>
</tr>
<tr>
<td>Cp</td>
<td>J/kg·K</td>
<td>Specific heat at constant pressure</td>
</tr>
<tr>
<td>Cms</td>
<td>W</td>
<td>Seal Moment Coefficient</td>
</tr>
<tr>
<td>Cw</td>
<td>m/s</td>
<td>Flow number</td>
</tr>
<tr>
<td>C wr R</td>
<td>m/s²</td>
<td>Rotor Surface Friction Coefficient (see 7.3)</td>
</tr>
<tr>
<td>Cifers</td>
<td>kW/m²</td>
<td>Stator Surface Friction Coefficient (see 7.3)</td>
</tr>
<tr>
<td>h</td>
<td>m</td>
<td>Seal fin height</td>
</tr>
<tr>
<td>H</td>
<td>kW</td>
<td>Calculated windage heating for seals with plain linings (power)</td>
</tr>
<tr>
<td>HHC</td>
<td>kW</td>
<td>Calculated windage heating for seals with honeycomb liners (power)</td>
</tr>
<tr>
<td>l</td>
<td>m</td>
<td>Fin pitch</td>
</tr>
<tr>
<td>m</td>
<td>kg/s</td>
<td>Leakage mass flow rate</td>
</tr>
<tr>
<td>N</td>
<td>RPM</td>
<td>Rotor speed</td>
</tr>
<tr>
<td>n</td>
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<td>Number of seal fins</td>
</tr>
<tr>
<td>P</td>
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<tr>
<td>r</td>
<td>m</td>
<td>Radius</td>
</tr>
<tr>
<td>Re</td>
<td></td>
<td>Modified Rotational Reynolds Number</td>
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<tr>
<td>Re</td>
<td></td>
<td>Rotational Reynolds Number</td>
</tr>
<tr>
<td>T</td>
<td>K</td>
<td>Air total temperature</td>
</tr>
<tr>
<td>t</td>
<td>K</td>
<td>Air static temperature</td>
</tr>
<tr>
<td>V core</td>
<td>m/s</td>
<td>Fluid core tangential velocity</td>
</tr>
<tr>
<td>W</td>
<td>kW</td>
<td>Measured windage heating (power)</td>
</tr>
<tr>
<td>X</td>
<td>m</td>
<td>Length of seal surface (rotor)</td>
</tr>
<tr>
<td>w R</td>
<td>rad/s</td>
<td>Rotor angular velocity</td>
</tr>
<tr>
<td>\rho</td>
<td>kg/m³</td>
<td>Air density</td>
</tr>
<tr>
<td>\mu</td>
<td>N.s/m²</td>
<td>Dynamic viscosity</td>
</tr>
<tr>
<td>\delta</td>
<td></td>
<td>Difference</td>
</tr>
<tr>
<td>\nu core</td>
<td>m/s</td>
<td>Fluid core velocity ratio</td>
</tr>
</tbody>
</table>

Suffices

R  Rotor
S  Stator
in  inlet
out  outlet

NOTE

FELTMENTAL is a seal lining material in sheet form made from randomly orientated fibres which are sintered to provide strength. It is produced by the Brunswick Corporation of the USA. The standard used in these tests was FM-515B.

Presented at the International Gas Turbine and Aeroengine Congress and Exposition
The Hague, Netherlands — June 13-16, 1994
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until September 30, 1994
1. INTRODUCTION

With improvements in efficiency of main gas path components of gas turbines (i.e. compressors and turbines) becoming ever more difficult and costly to achieve, attention continues to be directed towards reductions in secondary losses. Since the internal air system, though vital to satisfactory functioning of the engine, is entirely parasitic on the main cycle, increased effort is being put into its understanding and optimisation.

The internal air system of a gas turbine engine performs many complex, interrelated functions. The system extracts cooling air from the main gas path, and distributes it so as to provide a thermal environment that will maintain engine structural integrity and running clearance goals under steady state and transient conditions. At the same time it also pressurises (i.e. seals), cools and thrust balances the rotor.

The viscous drag on rotating components, the so called windage effect, represents both a direct loss of power from the cycle and an input of energy in the form of heat into the internal cooling system. The degraded quality of the cooling air in turn necessitates increased quantities of cooling flow (for such components as discs and blades) to be bled from the main annulus. The effects of windage on performance are therefore compounded.

Much information has been published on the windage of discs (e.g. Refs 2-8) and some, though much less information is available on the windage of cylinders, cones and protrusions (e.g. Refs 9, 10, 11, 12). At the time of starting the work described in this paper, no published information was available on the windage heating of air passing through labyrinth seals.

Subsequently McGreehan and Ko (1989) presented results of windage tests and compared experimental data with a simple friction factor model. Due to high heat transfer caused to some extent by the design of the experiment, the experimental results at low loads could not be adequately predicted by the simple friction factor model. Since the experiments reported in the present paper were all for flow levels much higher than for Ref 13 it was hoped that the simple friction factor model proposed in that reference would be able to match the measured data without the same problem of high heat loss at low flows being apparent.

The model of McGreehan & Ko, using the expressions for surface friction coefficients proposed in that paper, has been compared with the newly published experimental data for a range of leakage flows, rotor speeds and seal fin numbers. Results of some limited test cases using a full CFD analysis of the ‘straight through’ test seal are also compared with the experimental data. The objectives of the experiments reported in this paper were to establish the effect of leakage flow on seals windage power (the effect of flow on seals windage had previously not been allowed for in systems calculations) and to carry out a parametric study of the effect of fin geometry and seal lining materials on seals windage.

2.0 Description of Facility and Rig

The Rolls-Royce Internal Air Systems Research facility is installed on the Hucknall Thermal Site and has access to airflows up to 1.474 kg/sec at temperatures up to 650°C and pressures up to 4136 kPa. The part of the rig used for this programme of work (see Fig 1) is essentially a test box which houses cantilevered rotor assemblies of diameters up to 0.47 m. The rotor on test is driven by a 74.5 kW electric motor via a 10:1 ratio gearbox which allows speeds up to 13500 rpm to be achieved. The test unit is designed for maximum conditions of 345 kPa and 250°C.

The relevant operating non-dimensional parameters for the rig are very close to engine values as shown in Table 1.

### Table 1

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Engine</th>
<th>Rig Max</th>
</tr>
</thead>
<tbody>
<tr>
<td>$N_{AR}$</td>
<td>370</td>
<td>600</td>
</tr>
<tr>
<td>$M \times R^{1/4}$</td>
<td>$4.0 \times 10^4$</td>
<td>$8.0 \times 10^4$</td>
</tr>
<tr>
<td>$\frac{m}{L_{R}}$</td>
<td>$3.0 \times 10^7$</td>
<td>$1.7 \times 10^7$</td>
</tr>
</tbody>
</table>

For the labyrinth seal tests reported in this paper, the test air was fed via a control valve and an airflow measuring section into the Entry Chamber. The flow then splits to pass through the Test Seal and also the Rear Balance Seal, thereby cooling the pocket between those two seals. On leaving the Test Seal, the air splits again. Some of the air flows directly out of the Entry Chamber and the remainder flows through the Rear Balance Seal thereby ventilating the rear pocket and the rear of the disc before exiting from the Cooling Air Rear Compartment. The outlets from the Exit Chamber and Cooling Air Rear Compartment are both fitted with airflow measuring sections. The Front Cooling Air Compartment has a direct airfeed to supplement the air leakage over the Front Balance Seal. The mixed air then exits directly from the Front Cooling Air Compartment without the mass flow being measured.

3.0 Description of Test Seals

The test seals were basically of four types:

a. Straight through, radial fins
b. Straight through, inclined fins
c. Stepped, radial fins
d. 2 stage Brush

The important dimensions of the finned seals are shown in Fig 2.

All the finned seals were tested against plain stators. The stepped seal was also tested against a Felmetal lining, and some of the straight through seals were tested against honeycomb linings.

Table 2 shows the specification for each of the Build tested.
3.1 Determination of Running Clearance

With the end cover off the rig, the actual cold build radial clearances were measured by means of feeler gauges. Rotor growth checks had been carried out on previous runs of this rig using capacitance clearance probes and a high level of confidence had been established in prediction of both centrifugal and thermal growths. From the measured air temperatures within the running rig at any given rotor speed, the mean running clearance was calculated from the mean cold build clearance.

<table>
<thead>
<tr>
<th>TABLE 2</th>
<th>TEST SEALS DETAILS</th>
</tr>
</thead>
<tbody>
<tr>
<td>BUILD</td>
<td>TYPE</td>
</tr>
<tr>
<td>1</td>
<td>Stepped</td>
</tr>
<tr>
<td>2</td>
<td>Stepped + Feltmetal Lining</td>
</tr>
<tr>
<td>3</td>
<td>&quot;</td>
</tr>
<tr>
<td>4</td>
<td>Straight</td>
</tr>
<tr>
<td>5</td>
<td>&quot;</td>
</tr>
<tr>
<td>6</td>
<td>Straight + .062&quot; Cell Honeycomb</td>
</tr>
<tr>
<td>7</td>
<td>&quot;</td>
</tr>
<tr>
<td>8</td>
<td>Straight</td>
</tr>
<tr>
<td>9</td>
<td>Inclined Fins</td>
</tr>
<tr>
<td>10</td>
<td>Straight + Filled Honeycomb</td>
</tr>
<tr>
<td>11</td>
<td>Straight</td>
</tr>
<tr>
<td>12</td>
<td>Brush</td>
</tr>
<tr>
<td>13</td>
<td>Straight</td>
</tr>
<tr>
<td>14</td>
<td>Straight + .020&quot; Cell Honeycomb</td>
</tr>
<tr>
<td>15</td>
<td>Straight + .062&quot; Cell Honeycomb</td>
</tr>
</tbody>
</table>

4.0 Experimental Technique

The objective of the experimental work was to establish the effects of various geometric parameters and environmental conditions on the overall heat pickup of air passing through labyrinth seals. The experiment was conducted in a manner as possible all instrumentation was static (see 5.0 below) and intermediate parameters (e.g. in each seal cell) were not measured.

For an adiabatic system, all windage power absorbed in a rotor/stator system must be dissipated into the cooling air as heat. The experiments described in this paper were aimed at being adiabatic so that windage power absorbed by the air passing through the seal could be obtained very simply by means of measurement of enthalpy change in the main test cooling air i.e.:-

Windage Power absorbed = \( \Delta h \)

Within the labyrinth seal itself the flow can be considered as a series of throttling processes which in classical theory are normally assumed to be adiabatic because each throttling takes place in such a short length that the surface area across which heat can flow is very small. In the particular series of tests reported in this paper the temperature differences between the seal and its immediate environment were kept as low as possible to further reduce the potential heat loss.

The temperature rise of the air passing through the seal was obtained by measuring inlet and outlet temperatures immediately as the air entered and left the seal. Windage heat generated by the mounting disc rim section and the inlet and outlet balance seals was isolated from the test seal and main airflow by bleeding air into the front and rear seal cooling air compartments and thereby ventilating the section of disc rim radially spaced between the test seal and balance seals.

Any heat conducted from the hot side of the disc rim to the cold side would not affect the results since the upstream ventilation air carries this heat away from the inlet air thermocouples. Any heat conducted from the hot side of the seal stator to the cold side would not affect the results since this heat is dissipated into the main inlet air upstream of the inlet air thermocouples.

The outside of the rig was lagged and measurements were taken when the particular test set up had reached steady state.

5.0 Instrumentation

The location of the pressure and temperature tappings within the working section of the rig are shown in fig 3 below.
In order to improve accuracy, all main test parameter measuring points (pressures and temperatures) were triplicated and thermocouples used were of the exposed junction type. All thermocouples used were individually calibrated prior to fitment and so could be relied upon to read to an accuracy of ± 1°C.

The pitot/static measuring sections used were all calibrated prior to fitment against air standard meters and could be relied upon to facilitate flow measurement to an accuracy of ± 3%.

6.0 Test Conditions

Nominal inlet air temperature 25°C
Test seal inlet pressure level : 30, 40, 50 psia (ie = 207, 276, 345, kPa)
Rotor Speed : 0, 7, 9, 11, 13, Rpm \times 10^{-3}
Test seal pressure ratio 1.05 to 1.99
(i.e Pin/pout)

Normally three pressure ratios per build were tested. Wide clearance seals were tested towards the bottom end of the pressure ratio range and tight clearances towards the top end of the range. Where one seal configuration was tested at two clearances, the range of mass flow overlapped.

To limit the amount of testing some of the seals were tested at one inlet pressure only.

7. Discussion of results

The results from each test were plotted in the following forms:

(i) Flow function versus Pressure ratio
\[ \frac{\dot{m}}{\text{Pin}} \text{ versus } \frac{\text{Pin}}{\text{pout}} \]

(ii) Windage heating versus Rotor speed
\[ \dot{m} \text{ Cp} \delta T \text{ versus } \text{N} \]

7.1 Seal leakage

A typical set of flow characteristics is shown in fig 4. Since the flow results are not the main purpose of this paper, nothing more detailed is said of the results other than to say that no surprises were evident from the experiments. The order of merit was as expected from both previous two dimensional static tests and other rotating tests. The most efficient 'labyrinth' seal tested in terms of limiting leakage at a pressure ratio was the five finned stepped seal.

By plotting \( \frac{\dot{m}}{\text{Pin}} \) against pressure ratio, the result of the (zero clearance) brush seal could be compared with the positive clearance labyrinths. This result is shown in fig 5 and it can be seen that the sealing performance of the brush is best of all the seals tested with a leakage of only approximately 60% of the five finned stepped labyrinth running at 0.25 mm radial clearance.

Fig. 4

7.2 Windage Heating Measurements

A typical plot of measured windage heating versus rotor speed obtained from the experiments is shown in fig 7 below. The plot also includes curves of an early RR model for labyrinth seal windage based on simple free disc and free cylinder windage. Since the free disc and cylinder moment coefficients only vary with rotational Reynolds number, the model did not allow for variation in seal leakage flow (which is a function of pressure ratio). It can also be seen from the plot that whilst the model adequately predicted the variation of windage with pressure level and hence density, the predicted variation with rpm is distinctly different from that observed. It was in fact due to the known inadequacies of this model in allowing for seal flow that the experimental programme had been initiated.
By plotting measured windage heating against leakage flow for all test builds at a speed and a pressure level, it is possible to make certain observations directly. A typical plot for the 13,000 rpm speed and an inlet pressure of 344.7 kPa is shown in fig 8 below.

WINDAGE HEATING V RPM

RUNNING CLEARANCE (mm NOM.)
0.25 0.75
+ INLET PRESSURE = 344.7 kPa
- INLET PRESSURE = 275.8 kPa
* CALCULATED BY
** EARLY RP MODEL

Fig. 7

The observations which are possible from plotting in this form are:

(a) A clear relationship exists between windage heating and leakage flow largely independent of clearance over the range tested. Where the same seal was tested at more than one clearance, the results fall close to a common line.

(b) Both inclined fin seals and stepped seals generate similar levels of heat to the straight through seal at the same airflow (but the flows through the inclined fin seal and stepped seal are of course less than the straight through seal at the same pressure ratio).

(c) As might have been anticipated, the brush seal results fall on the extrapolated single fin seal line but, at any pressure ratio, the flow through the brush was obviously very small by comparison.

(d) At a given flow level, honeycomb linings result in an increase in measured windage of approximately 15%.

(e) For the only direct 'back to back' tests carried out on the two sizes of honeycomb lining (tested against a large pitch 3 fin seal), the measured windage was the same for each at a given level of seal leakage flow.

From all the windage measurements made on the various seals with plain linings, the simple correlation for seal windage given below was determined with no great difficulty.

\[ H = C_{ms} \times n \times \left( \frac{P}{n} \right) \]

Where \( C_{ms} \) is a seal moment coefficient given by

-0.06 \[ \frac{C}{n} \]

0.55 \[ \frac{C}{n} \]

0.65

It is found that the correlation predicts the measured seal windage data for the plain liners to within + 25% at all levels of flow. The few points which fall outside these limits all overestimate windage (the single worst value being + 40%). Windage heat for the honeycomb lined seals can be predicted to similar levels of accuracy using

KHC = 1.15 H

7.3 Comparison between measured seal windage and values computed using the McGreehan and Ko model suggested in 89-GT-220

The attraction of this model over the above correlation is the fact that it is based on a relatively simple but plausible assumption about the actual physics of seal windage heating. This method of predicting seal windage, if sufficiently accurate, is a much quicker means for use in airflow analysis than a full CFD analysis of the flow within the seal.

Results are presented for the model using the same surface friction coefficients used in 89-GT-220.

\[ C_{FR} = 0.042 \left[ 1 - \beta \right]^{1.35} Re^{0.2} \]

and

\[ C_{FS} = 0.053 \beta^{1.87} Re^{0.2} \]

As can be seen from fig 9 below, the curves produced by the model are of the same form as seen for the experimental results in fig 8. Windage is seen to fall to low levels at low leakage flows and to approach a maximum at high flows as the seal becomes choked. Computed curves are shown for 3 and 5 fin seals for inlet swirl fractions of 0 and 0.1. The experimental values for the 5 fin seal are well predicted by the model between the set limits of inlet swirl fraction. The results for the 3 fin seal are not as well predicted but are only significantly underpredicted at the highest flows.

Fig. 8 Typical experimental results

Fig. 9 Straight seals - plain liners
Typical carpet plots for the five fin straight seal generated using the model are given in figs 10 and 11 for rotor speeds of 13,000 and 10,000 rpm respectively. The main carpets are generated assuming an inlet swirl fraction of zero and a half seal cell at inlet and exit. It can be seen that the model predicts the observed effect that actual clearance has little effect on windage until the seal approaches choking (for constant upstream pressure). The major effect of clearance change at a pressure ratio is a change of flow which does have a direct and significant effect on windage. The model, run with zero inlet swirl, is seen to slightly overpredict the measured data. However it can be seen that application of even a small amount of inlet swirl (in this case to 0.11) encompasses most of the test points shown. It is quite possible that whilst every effort was made to run an adiabatic experiment with zero inlet swirl, a small level of swirl could have been generated in the entry chamber or some small heat loss could have occurred. Either effect could result in the small discrepancy between the model and the experimental results.

![Carpet plot](image)

**Fig. 10 Carpet plot of (N and W) versus m and C (5 Fin straight seal - plain liners, N = 13000 Rpm)**

**Fig. 11 Carpet plot of (N and W) versus m and C (5 Fin straight seal - plain liners, N = 10000 Rpm)**

7.4 Comparison between measured seal windage and values computed using the RR CFD code ‘PACE’

A very limited investigation was carried out to see how well the experimental results could be matched using an existing RR CFD code (‘PACE’). The CFD method used is a finite volume Navier–Stokes solver which employs boundary-fitted coordinates and uses the k-ε turbulence model with alternative near wall treatments. In this case the k-ε region was coupled to the near wall regions using a near-wall low Reynolds number one-equation k-ε model. A full description of the method is given in Ref. 14.

Since the method is strictly for incompressible flow, the experimental points chosen for comparison were at the two lowest pressure ratios tested on the 5 fin straight seal operating at the largest clearance. An inlet swirl fraction of 0 was used and adiabatic conditions were assumed. The results of the CFD computation for the two different flows (both for rotor speeds of 13000 rpm) are given below. It can be seen that the agreement with the experimental results is very good.

<table>
<thead>
<tr>
<th>Case</th>
<th>RPM</th>
<th>P.R.</th>
<th>m</th>
<th>Pin(kPa)</th>
<th>measured W(kW)</th>
<th>Computed W(kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>13000</td>
<td>1.1</td>
<td>0.217</td>
<td>5</td>
<td>344.7</td>
<td>4.178</td>
</tr>
<tr>
<td>2</td>
<td>10000</td>
<td>1.3</td>
<td>0.421</td>
<td>5</td>
<td>344.7</td>
<td>5.412</td>
</tr>
</tbody>
</table>

It is encouraging to note that accurate predictions of seal windage for sealing operating 'incompressibly' are possible using CFD techniques if a particular case justifies the effort. A further extension of the work would be to use CFD to further calibrate the simple friction factor method.

8. CONCLUSIONS

The main conclusions from the work presented can be summarized as follows:-

8.1 The power absorbed due to frictional drag in labyrinth seals is strongly affected by the airflow (m).

8.2 Different designs of labyrinth seals with the same nominal radii and surface areas (which may have different sealing efficiencies) will result in similar levels of windage heating at a given flow level.

8.3 Labyrinth seal windage heating for a given inlet pressure will reach a maximum before the seal is choked. This is due to falling pressure levels in the seal cells outweighing the less rapid increase in flow as choking is approached.

8.4 Honeycomb linings of two different cell sizes were both found to increase windage by approximately 15%.

8.5 Feltmetal lined seals resulted in even higher windage heating than honeycomb lined seals. (The 'step gaps' present on the feltmetal seal but absent on the honeycomb seal in the present tests could possibly have been the cause of this effect due to recirculation of flow).

8.6 Windage heating for the brush seal was of a very low level - due no doubt to the very low leakage flows.

8.7 A simple correlation can be used to predict windage heating in plain lined labyrinth seals to approximately ± 25%.

8.8 A better, more accurate and still simple method of predicting seal windage is that proposed by McGreehan and Ko (1989).

8.9 The most accurate method for predicting labyrinth seal windage was found to be by the use of the RR 'PACE' finite volume Navier Stokes solver. This method however requires much more effort to execute and may not be justified for normal 'systems' calculations.
9. ACKNOWLEDGEMENTS

The authors would like to thank the UK Ministry of Defence Procurement Executive for providing part funding for the work presented in this paper and would also like to thank Rolls-Royce PLC for allowing publication of the work. Thanks also goes to G P Virr who carried out the CFD calculations referred to in section 7.4.

10. REFERENCES

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