Rim Sealing of Rotor-Stator Wheelspaces in the Absence of External Flow

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ABSTRACT
Sealing of the cavity formed between a rotating disc and a stator in the absence of a forced external flow is considered. In these circumstances the pumping action of the rotating disc may draw fluid into the cavity through the rim seal. Minimum cavity throughflow rates required to prevent such ingress are estimated experimentally and from a mathematical model. The results are compared with other workers' measurements. Measurements for three different types of rim seal are reported for a range of seal clearances and for rotational Reynolds numbers up to $3 \times 10^6$. The mathematical model is found to correlate the experimental data reasonably well.

NOMENCLATURE

- $C_d$: discharge coefficient
- $C_w = \dot{m}/\mu R_o$: mass flow parameter
- $G = s/R_o$: cavity aspect ratio
- $G_C = s_c/R_o$: seal clearance ratio
- $k$: empirical constant in seal model
- $\dot{m}$: coolant mass flow rate
- $\dot{m}_{in}$: ingress mass flow rate
- $r$: radial coordinate
- $R_o$: outer radius of disc
- $Re_0 = \rho R_0^2/\mu$: rotational Reynolds number
- $s$: axial gap between rotor and stator
- $s_c$: seal clearance
- $u_{dm}$: mixed out radial velocity for near-wall flow on the rotating disc at $x = 1$
- $u_m$: mean velocity through the seal in r-z plane
- $x = r/R_o$: non-dimensional radius
- $z$: axial coordinate
- $\Delta p$: pressure difference across seal
- $\Delta p_C$: pressure difference in cavity from $x = 0.95$ to $x = 1$
- $\lambda = C_w/Re_0^0.6$: throughflow parameter
- $\mu$: fluid viscosity
- $\rho$: fluid density
- $\Phi = \dot{m}/(\dot{m} + \dot{m}_{in})$: cooling effectiveness
- $\omega$: angular speed of rotor
- $\text{subscript min}$: minimum value to seal cavity

1. INTRODUCTION

In an earlier paper (Chew, 1989) a mathematical model was proposed for ingestion of external flow into the cavity formed between a rotating disc and a stationary shroud. Some encouraging agreement with the available experimental data was found, but several uncertainties remained owing to apparent inaccuracies in the available data. In the present
contribution some new experimental data is presented and the mathematical model is re-examined.

Rim sealing has been investigated by several groups of workers in recent years and it is now generally accepted that the presence of an external flow, simulating the main gas-path in an engine, can severely modify or dominate the sealing behaviour. In particular, circumferential pressure asymmetries in the external flow are thought to be important. Nevertheless attention here is confined to "rotationally-dominated" conditions in which ingress is driven by the pumping action of the rotating disc rather than pressure asymmetries in the external flow. This is a useful test case for modelling and, despite being a simplification of engine conditions, is a severe test of our understanding of the problem. It is also worth noting that there are engine configurations, such as the inner seal in a double-sealed arrangement, where a rotationally-dominated model might be expected to apply.

Measurements were obtained from the experimental rig described in a companion paper (Dadkhah et al, 1991) and in more detail by Dadkhah (1990). The outer radius of the rotor in this rig is 200 mm. Apart from the region of the seal the rotor has a plane surface and faces a plane stator, spaced 20 mm axially from the rotor. Sealing flow entered the cavity through a central hole in the stator. The rig has the facility for external flow, with and without swirl, and the measurement techniques applied include static pressure tappings, gas concentrations and total pressure probes. However, attention here is confined to the static pressure measurements for the case when there is no external flow. Three different seals were tested: a simple axial clearance seal and two radial seals. Details of these seals are given in Fig.1.

For completeness a brief description of the mathematical model will be given in the next section. Experimental and theoretical results are then presented and compared in the following two sections. A further comparison of the model with the data of Daniels et al (1990) is then given in section 5, before the conclusions from this study are summarised in section 6.

2. THE MATHEMATICAL MODEL

The approach adopted is to couple a solution for the flow in the wheelspace to a model for the seal flow. A momentum-integral method was chosen to calculate the cavity flow because of its computational efficiency compared to numerical solutions of the governing partial differential equations. For further discussion of this model the reader is referred to Chew (1989, 1990). The main features of the model are described below.

2.1 The Cavity Flow

Separate turbulent boundary layers are assumed to form on the rotating and stationary discs and on the cylindrical outer shroud surface. In the inviscid region between the boundary layers a rotating core, in which the radial velocity component vanishes, forms in the outer part of the wheelspace. Inlet conditions affect the flow towards the centre of the cavity, particularly in the inviscid region, but in the rotationally-dominated case, the flow in the outer part of the cavity is insensitive to the inlet configuration.

The momentum-integral solution for the boundary layer on the rotating-disc involves a direct extension of von Karman's (1921) method for the free disc, allowing for fluid rotation outside the boundary layer. A modified von Karman type treatment is applied on the stationary disc, with the radial momentum equation being dropped and a limiting flow angle at the wall being assumed. On the cylindrical surface a constant mass flow boundary layer and a constant friction factor governing the circumferential shear stress is assumed. The ordinary differential equations for the boundary layer flows are coupled through solutions for the inviscid flow regions. These are the free vortex relations in the source region, and the rotating core solution in which there is no radial motion and axial gradients are zero. The boundary between the source and the rotating core regions is taken as the radius at which all the supplied flow is entrained into the boundary layer on the rotor. Although the model does allow for the inclusion of a specified ingestion flow, this option is not used in the results given here.

Numerical methods are used to obtain solutions to the system of equations described above. For the essentially incompressible conditions considered here, the solution, written in a suitable non-dimensional form, is a function of the non-dimensional radius \( x = r/R_o \), the aspect ratio of the cavity \( G = s/R_o \), and a non-dimensional throughflow coefficient \( \lambda \). In terms of the rotational Reynolds number \( Re_o = \omega R_o^2 / \nu \) and the mass flow coefficient \( \alpha = \dot{m} / (\omega R_o^2) \), \( \lambda \) is defined as

\[
\lambda = \frac{\alpha}{Re_o^{0.8}}.
\]

As described in earlier references this model has been validated against moment coefficient data for rotor-stator systems. A comparison with core velocity measurements (Dadkhah et al, 1991) indicates that the model predicts core velocities somewhat lower than measurements. However, at low...
values of $\lambda$ the model agrees reasonably well with measurements.

2.2 The Seal Model

Consider first the case when the supplied throughflow is greater than the minimum rate required to prevent ingress to the cavity ($C_w = C_{w,\text{min}}$). It may be argued from order of magnitude considerations that the effect of rotation on the pressure drop across the seal will be principally through changes induced in the radial and axial pressure drop across the seal. The fluid flowing out of the seal first picks up a radial velocity component as it is entrained into the boundary layer on the rotor. To quantify this effect a mixed-out radial velocity ($u_{dm}$) is calculated from the integral solution at the rotor outer radius. Assuming that the seal flow is supplied from the near wall region of this layer, $u_{dm}$ is the velocity of a uniform stream that would have the same radial momentum as the flow approaching the seal. It is then postulated (by analogy with pipe orifice flow) that the pressure drop across the seal may be represented by the equation

$$\Delta p = \frac{1}{2} \frac{C_d}{k^2} \left(u^2 \frac{m}{w_R} + k^2 u_{dm}^2\right)$$

where $C_d$ is a discharge coefficient, $u_m$ is the mean velocity in the $r-z$ plane through the seal, and $k$ is a constant, the value of which depends on the seal type and is determined by matching to experimental results.

Assuming that the cavity is just sealed when $\Delta p = 0$, it follows from eqn.(1) that this occurs when $u_m = k u_{dm}$. Calculating $u_m$ from the boundary layer at $x = 1$ (from analogy with pipe orifice flow) and the throughflow rate $m$, and rearranging gives

$$C_{w,\text{min}} = 2\pi k (\frac{u_{dm}}{w_R}) Q_c Re_p$$

For a given cavity geometry ($u_{dm}/w_R$) will be a function of $\lambda$ only. It is interesting to note that for the axial seal in fig. 1 $k$ would be expected to be of order unity (from analogy with pipe orifice flow) and ($u_{dm}/w_R$) is expected to be of order 0.1 (from von Karman's free disc solution). Putting these figures in eqn.(2) gives $C_{w,\text{min}} = 0.628 Q_c Re_p$. This is close to Bayley and Owen's (1970) empirical correlation for the minimum sealing flow for this configuration which gives $C_{w,\text{min}} = 0.61 Q_c Re_p$. In terms of the throughflow parameter $\lambda$, eqn.(2) gives

$$\lambda_{\text{min}} = 2\pi k (\frac{u_{dm}}{w_R}) (Q_c Re_p^{-0.2})$$

Since ($u_{dm}/w_R$) depends only on $\lambda$, it follows that for a given cavity geometry and seal type $\lambda_{\text{min}}$ will be a function of ($Q_c Re_p^{-0.2}$) only.

In the experimental work it was impractical to measure the cavity pressure at the outer radial position $x = 1$ for all seals. For consistency across all seal types the estimates of the pressure differences across the seal were based on the cavity pressure at $x = 0.95$. It will be of interest to compare the minimum sealing flows given by the model using the pressure at $x = 0.95$ with those obtained from eqn.(3), and with those estimated from the measurements. Denoting the pressure rise in the cavity between $x = 0.95$ and $x = 1$ as $\Delta p$, it follows from eqn.(1) that estimating the minimum sealing flow from the pressure at $x = 0.95$ gives

$$\lambda_{\text{min}} = 2\pi Q_c Re_p^{0.2} \left[\frac{u^2}{w_R^2} + \frac{k^2}{Q_c Re_p^{0.2}}\right]^{0.5}$$

As expected, for small $\Delta p$ this eqn approaches eqn.(3). Eqn.(4) again indicates that for a particular configuration $\lambda_{\text{min}}$ will be a function of $Q_c Re_p^{-0.2}$ only.

When $\lambda$ is less than $\lambda_{\text{min}}$, ingress occurs. A useful measure of the ingress level is the sealing effectiveness $\Phi$ defined as $m/(m + m_1)$ where $m_1$ is the ingress mass flow rate. Based on estimates of $\Phi$ from concentration measurements, Graber et al (1987) suggested the exponential formula $\Phi = 1 - e^{-\alpha x}$, where $\alpha$ is a constant for a particular seal clearance. Experimental data from the present rig support this relationship (see Dadkhah et al, 1991). An alternative curve fit to the measured values of $\Phi$ is obtained by assuming that the ingress flow rate is $20\%$ of $(\lambda_{\text{min}} - \lambda)$. This gives

$$\Phi = \left(0.8\lambda + 0.2\lambda_{\text{min}}\right)$$

It may be shown that if $\lambda_{\text{min}}$ is taken as the value of $\lambda$ for which the exponential relationship gives $\Phi = 0.98$, the maximum difference in $\Phi$ given by the two formulae is 0.35 and this is within the scatter of the data used for the curve fit. In the present model of ingress eqn.(5) is used with $\lambda_{\text{min}}$ estimated from eqn.(3) above.

![Fig.2 Discharge coefficients for the three seals.](attachment:image.png)

3. THE AXIAL CLEARANCE SEAL

This seal has previously been studied experimentally by Bayley and Owen (1970), Owen and Phadke (1980) and Phadke and Owen (1982, 1983). As in the present study the minimum sealing flow has been estimated from pressure measurements. Bayley and Owen, and Owen and Phadke used pressure tappings on the cylindrical shroud (at $x = 1$) in estimating
Phadke and Owen used tappings on the stator at
\( x = 0.975 \). In the present study a pressure tapping at
\( x = 0.95 \) was used. All workers have assumed that the
cavity is just sealed when \( \Delta p = 0 \). Use of this
criterion is supported by comparison with flow visualisation in the papers by Owen and Phadke and by comparison with concentration measurements in Phadke and Owen (1983).

Before considering the effects of rotation it is
useful to examine the seal behaviour for \( \text{Re}_g = 0 \). Fig.2 shows discharge coefficients calculated from the measured pressure drop across the seal at different seal clearances. The error bands on these
data show the sensitivity to an error of 0.0254mm
(0.001in) in the seal clearance; this was considered to be representative of the uncertainty in these experiments (Sadkha, 1989). As might be expected the discharge coefficients for this seal agree fairly well with those for the annular orifice between a circular disc and cylindrical tube. Annular orifice results were obtained from Bell and Bergelin's (1957) correlation and are shown as a line on the figure. the present results confirm difficulties with the Phadke and Owen data, which differ widely from Bell and Bergelin's results at smaller clearances (see Chew, 1989).

A comparison of predicted and measured minimum sealing flows is shown in Fig.3. Results are shown for two values of \( \text{Re}_g \), \( 9 \times 10^5 \) and \( 3.28 \times 10^6 \), representing the extremes of the experimental range. The constant \( k \) in the model was set to 0.8 for this
seal. This value was found in earlier studies to be representative of the uncertainty in these
measurements using the differential pressure
criterion for incipient ingestion. Von Karman's
(1921) solution for free disc entrainment is also included as this gives a guide to the expected level of \( C_{w,min} \) at high values of \( G_c \). The correlation of the present data was obtained assuming a relation of the form \( C_{w,min} \propto G_c^{0.6} \text{Re}_g^{0.4} \); a good fit to experimental data was found. Bayley and Owen's correlation was based on data for \( G_c = 0.0033 \) and 0.0067; the discrepancies of this correlation with the present results and those of Phadke and Owen at \( G_c = 0.04 \) is not surprising. For \( G_c = 0.01 \) the present results agree well with those of Phadke and Owen and for \( G_c \geq 0.04 \) these also agree with Bayley and Owen's correlation. There are significant discrepancies between the three correlations at lower values of \( G_c \). At \( G_c = 0.0025 \) Phadke and Owen's
cavity pressure at \( x = 0.95 \) agree to within a few
percent. At \( G_c = 0.002 \) the discrepancy is of order
20%. This may partly be explained by the fact that \( C_d = 0.65 \) in this calculation which somewhat underestimates the discharge coefficient at the smaller clearance ratios. It may be seen from eqn.(4) that increasing \( C_d \) will lead to a higher prediction of \( C_{w,min} \). The difference between the predictions based on pressure at \( x = 0.95 \) and that at \( x = 1 \) is proportionally higher at lower values of \( G_c \). This is to be expected as \( C_{w,min} \) is reduced at low clearances and the non-dimensional pressure differential in the cavity (\( 2\Delta p_c/p_w R^2 \)) will be higher due to increased fluid rotation.

Fig.4 gives a comparison of various correlations for \( C_{w,min} \) all of which are based on measurements using the differential pressure
criterion for incipient ingestion. Von Karman's
(1921) solution for free disc entrainment is also included as this gives a guide to the expected level of \( C_{w,min} \) at high values of \( G_c \). The correlation of the present data was obtained assuming a relation of the form \( C_{w,min} \propto G_c^{0.6} \text{Re}_g^{0.4} \); a good fit to experimental data was found. Bayley and Owen's correlation was based on data for \( G_c = 0.0033 \) and 0.0067; the discrepancies of this correlation with the present results and those of Phadke and Owen at \( G_c = 0.04 \) is not surprising. For \( G_c = 0.01 \) the present results agree well with those of Phadke and Owen and for \( G_c \geq 0.04 \) these also agree with Bayley and Owen's correlation. There are significant discrepancies between the three correlations at lower values of \( G_c \). At \( G_c = 0.0025 \) Phadke and Owen's
result is 80% higher than that of Bayley and Owen, with the present correlation falling between the two. Considering that the present results are expected to give high values of \( C_{w\min} \) owing to the use of the pressure measurement at \( x = 0.95 \), and that the accuracy of the clearance setting in Phadke and Owen's experiments is questionable, Bayley and Owen's correlation should perhaps be preferred at this gap ratio. Overall this comparison supports the use of the present model which agrees with Bayley and Owen's result at small clearances and with Phadke and Owen's results at larger clearances.

4. TWO RADIAL CLEARANCE SEALS

The two radial clearance seals considered are shown in fig.1. Seal A has a lip (width 0.5 mm) on the rotor which is overlapped by the stator. With this arrangement the flow pumped outwards in the rotor boundary layer will impinge on the stator and be turned before passing through the seal. Seal B has a similar lip on the stator which is overlapped by the rotor. In this case the shape of the rotor will tend to direct the pumped flow into the seal. As for the axial clearance seal, estimates of the minimum sealing flow were obtained from the differential pressure criterion, using the pressure measurement from an internal tapping at \( x = 0.95 \) in the stator and external tappings in the stator shroud some distance away from the seal.

Discharge coefficients for these seals for non-rotating conditions are shown in fig.2. These are somewhat lower than the values for the axial clearance seal. This may be due to the small length-to-clearance ratio and the slightly more tortuous path afforded to the air with these seals. Note also that for seal A with \( G_{c} = 0.01 \) the clearance between the tapered surface of the shroud and the corner of the lip is approaching the radial seal clearance. This may account for the surprisingly low discharge coefficient (0.48) in this case.

Experimental and theoretical results for the minimum sealing flow with seal A are shown in fig.5. In the model \( k \) has been set to 0.4. This value approximately correlates Phadke and Owen's (1982, 1983) large clearance results for a simple radial seal in which the stator shroud overlaps the rotor (see fig.7(v)). For the predictions based on pressure at \( x = 0.95 \) values of \( C_{d} \) of 0.5 and 0.65 have been used. It is apparent that for this seal, which performs somewhat better that the axial clearance seal, the predicted minimum sealing flows are more sensitive to the choice of pressure used in estimating \( \lambda_{\min} \). At the lower value of \( G_{c} \) considered (= 0.002) this effect gives almost a 100% increase in \( \lambda_{\min} \) when the pressure at \( x = 0.95 \) is used as opposed to the pressure at \( x = 1.0 \). As for the axial clearance seal, the model appears to underpredict the minimum sealing flow at the small clearance ratio, but the measurements for \( G_{c} = 0.006 \) (grouped around \( G_{c} = 0.01 \)) are in good agreement with the theory. For \( G_{c} = 0.01 \) the theory overestimates the minimum sealing flow by about 20%. This departure may be related to the rather low discharge coefficient observed in fig.2 and discussed above. Obviously some caution is advisable in interpreting these results as the assumptions underlying the model are subject to some uncertainty. However, this comparison does give further support for the validity of the present analysis in situations where the seal geometry is relatively simple.

Minimum sealing flows for seal B are shown in fig.6. The performance of this seal is similar to that of the axial clearance seal and so a value of \( k = 0.8 \) has been used in the model. The experimental results in this figure are for nominal values of \( G_{c} \) of 0.002, 0.006 and 0.01. Bayley and Owen's (1970) correlation for the axial clearance seal is also
shown on this graph. Agreement between theory and experiment is considered satisfactory, although there is some notable scatter. Note, that in the model, the value for $C_d$ of 0.65 is possibly rather high, thus overestimating $\lambda_m$ based on the pressure at $x = 0.95$.

5. OTHER SEAL GEOMETRIES AND RATES OF INGESTION

The various seal geometries that have been considered with the present mathematical model are illustrated in fig.7. Values of the model constant $k$, associated with each seal are also given in the figure. Cases (i), (iii) and (vi) are the seals considered above. Seals (iii) and (iv) were discussed by Chew (1989) and seal (iv) has been shown by Daniels et al (1990) to perform similarly to seal (iii). Owing to uncertainties and incompleteness in the experimental data available the model cannot be said to have been completely validated for all these seals, but it does offer a reasonable method of estimating the performance of the seals.

The model constant $k$ provides a measure of the seal performance. Poor seal types have high values of $k$ and good seal types have low values. Examining the seal geometries in fig.7 and considering the likely flow paths some explanations for the behaviour of the various seals is possible. It should be remembered that fluid will be pumped radially outwards in the boundary layer on the rotor, and that this fluid has a relatively high angular velocity and so will tend to centrifuge outwards. For seal (i) the pumped flow feeds directly into the seal, resulting in relatively poor seal performance; this is also the case for seal (iii). With seals (iii) and (iv) the disc pumping flow must turn and cross the cavity before reaching the seal; this introduces some extra resistance to the pumped flow and results in an improvement in seal performance. It might be thought that for seal (iii) the pumped flow must impinge on the stator and be turned again before reaching the seal. However, owing to the centrifuging effect, there will be little resistance to this additional turn. In the two best seals, (v) and (vi), the pumped flow will impinge on the stator and must turn sharply before entering the seal.

Egn.(5) for the sealing effectiveness can only be regarded as a guide to the rate of ingestion when the supplied flow is less than the minimum sealing flow. Because mixing of the ingested and supplied flow within the cavity will not be complete, experimental estimates of $\Phi$ from concentration measurements will depend somewhat on the choice of sampling positions. Prediction of the distribution of the ingested gas within the cavity is beyond the capabilities of the current model. Nevertheless this model may be useful in supplying quick estimates of the sealing effectiveness for engineering calculations. Application of the method to seal type (iv) in fig.7 and comparison with Daniels et al measurements of sealing effectiveness is shown in fig.8. (It is reasonable to apply the rotationally-dominated theory to this case as the axial velocity of the external flow in those experiments was small.) Estimates of $\lambda_{\text{min}}$ were obtained from the above model (with $k = 0.65$) and eqn.(5) was then used to calculate $\Phi$. For $Q_c = 0.0048$ the agreement of the model with experiment is very good. For $Q_c = 0.0024$ the agreement is not so good but, in view of the uncertainty level due to sampling position and other factors, the model may be considered to give a reasonable estimate of $\Phi$.

6. CONCLUSIONS

Experimental measurements of the minimum throughflow rate required to prevent ingestion into a shrouded disc system have been presented for three different types of seal. These results were compared with predictions from a mathematical model.
and with other workers' measurements. A differential pressure criterion has been used to detect incipient ingress; this assumes that the cavity is just sealed when the pressure differences across the seal is zero. Analysis showed that for small seal clearances this criterion is sensitive to the position of the cavity pressure measurement owing to the centripetal pressure gradient in the cavity. When this effect was taken into account reasonable agreement between the mathematical model and the measurements was found.

Axial clearance seal results were compared with earlier correlations of measurements by Bayley and Owen (1970) and Phadke and Owen (1983). All three correlations agree at a gap ratio \( G_c \) of 0.01. At higher values of \( G_c \), which are outside Bayley and Owen's experimental range, the present results agree with those of Phadke and Owen. At lower values of \( G_c \) the three correlations diverge. Differences between the present correlation and that of Bayley and Owen can largely be explained by the effect of pressure measurement position mentioned above. Differences with Phadke and Owen's results seem to confirm inaccuracies in their experiments at small seal clearances.

Two radial clearance seals were tested. Seal A, with a lip on the rotor and an overlapping stator was the most effective. For this seal the effect of pressure measurement position was relatively larger and so the results were less conclusive. Nevertheless some encouraging agreement with the mathematical model was found. The performance of Seal B, with a lip on the stator and overlapping rotor, was similar to the axial clearance seal and, again, reasonable agreement with the mathematical model was found.

While the rather elementary nature and limitations of the present model must be acknowledged, it does apparently correlate the available data for a number of different seals and it is to be preferred to earlier empirical correlations. Together with the assumption that for \( C_w < C_{w,\text{min}} \) the ingress rate is 20% of \( (C_{w,\text{min}} - C_w) \) the model can be used to obtain a first estimate of the sealing effectiveness. This relation has been shown to give a reasonable fit to Daniels et al.'s (1990) estimates of sealing effectiveness based on concentration measurements. Clearly much uncertainty remains and further development of the model is needed to account for the influence of external flow.

In this continuing research programme further experimental, modelling and computational fluid dynamics studies are planned.

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