A Dynamic Analysis of Rotary Combustion Engine Seals

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ABSTRACT

Real time work cell pressures are incorporated into a dynamic analysis of the gas sealing grid in Rotary Combustion Engines. The analysis which utilizes only first principal concepts accounts for apex seal separation from the trochoidal bore, apex seal shifting between the sides of its restraining channel, and apex seal rotation within the restraining channel. The results predict that apex seals do separate from the trochoidal bore and shift between the sides of their channels. The results also show that these two motions are regularly initiated by a seal rotation. The predicted motion of the apex seals compares favorably with experimental results. Frictional losses associated with the sealing grid are also calculated and compare well with measurements obtained in a similar engine. A comparison of frictional losses when using steel and carbon apex seals has also been made as well as friction losses for single and dual side sealing.

BEFORE STARTING A DISCUSSION of the factors involved with sealing a Rotary Combustion Engine (RCE) it is helpful to examine the standard network of seals used in RCEs. Referring to Fig. 1 which was taken from Ansdale (1)*, one sees the long curved side seals which contact the end covers of the cylinder and three apex seals which separate the three working cells of the engine. Examination of this sealing network allows appreciation of the additional difficulty involved in sealing RCEs. The long length of the side seals and the irregular path traversed by the apex seals present problems different from those encountered in sealing reciprocating piston engines. In this paper, a theoretical dynamic analysis of the forces acting on the seals, is coupled with experimentally obtained cell pressures to obtain a friction model of the gas sealing system. Both the possible separation of the apex seals from the trochoidal bore and the frictional losses associated with this type of sealing configuration will be examined.

Up to the present time, documentation of the forces acting on these seals has not appeared in the literature to any appreciable extent. Jones (2) shows an approximate breakdown of friction losses at 6000 rpm in a Curtiss-Wright RCI-60 engine. Here seal friction losses are reported to be about 1 1/2 times the losses in the bearings and gears. Yamamoto (3) shows gas sealing losses to be over 50 percent of the total normalized frictional losses in an unspecified rotary engine. To the author's knowledge, no prediction or measurement of the friction associated with each sealing component has been previously published.

Loss of contact between the apex seal and the trochoidal bore has been discussed by a number of researchers. Eberle and Klop (4), have cited leakage past the apex seal as a possible cause of higher hydrocarbon emissions and increased specific fuel consumption. Prasse et al. (5) and Rodgers et al. (6), have related seal separation to chatter marks which form on the trochoidal bore creating wear problems. Matsuurra et al. (7), experimentally have shown that separation of the apex seal from the bore does take place and have correlated the locations of these separations with low, non-negative, contact forces. Both Prasse and Matsuurra use Ansdale's typical cell pressure profile and inertial force relationship. Since a component of the contact force acting between an apex seal and the trochoidal bore arises from the pressure differential across the apex seal, an important addition to the past analyses is the inclusion of the actual differential pressure data obtained under operating conditions.

EXPERIMENTAL PRESSURE MEASUREMENT

The experimental measurement system used to obtain cell pressures during both motoring and

Knoll, Vilmann, Schock, and Stumpf
firing conditions was developed by NASA. Since this system has been discussed in detail in a previous paper by Schock et al. (8), this brief description will be presented here. Four piezoelectric pressure transducers were mounted in a turbocharged 1978 Mazda 12A nonemission type (no catalytic converter) engine. The locations of these transducers are shown in Fig. 2. Four transducers were used in order that a continuous pressure versus crank angle plot from a single system of interest could be recorded. Whenever the rotor occupied a position where two transducers were active, the average difference between these two transducers was calculated and used to offset the trailing transducer's output. A signal correlator was also designed to reduce the data resulting from these four transducers to that of the pressure acting in one cell during a complete cycle. Pressures were sampled at 2048 equally spaced angular positions throughout the 1080° of mainshaft rotation.

Pressure versus crank angle plots for all the operating conditions examined have a shape similar to that shown in Fig. 3. These plots depict the pressure acting in one working cell throughout 1080° of crank angle rotation. The 0° crank angle reference for these plots corresponds to the location where the system of interest under consideration contains a minimum volume and intake begins. The significant differences between the traces taken at the various operating conditions tested are the magnitude peak pressure near top-deadcenter (TDC) and the value of the intake baseline pressure. Values for the intake manifold pressure ranged from 37 kPa at 2939 rpm under motoring conditions to 351.6 kPa at 6000 rpm while turbocharged firing. The compression-firing peak pressures ranged from 682.6 kPa at 2939 rpm motoring to 3061.4 kPa at 6000 rpm firing. A summary of the data collected and the conditions under which that data was collected is shown in Table 1.

APEX SEAL LIFT OFF

An examination of the apex sealing configuration revealed that there are three possible movements of the apex seal with respect to its channel which could initiate separation. The seal could: (1) slide away from the bore while retaining contact with one of the channel's sides; (2) the seal could shift between the leading and trailing edges of its channel; and (3) the seal could pivot about one of its lower edges. Further investigation of the sliding movement separation would automatically occur. The other two initiating movements could cause a redistribution of the forces acting on the seal resulting in the seal beginning to slide away from the bore.

In order to examine the possibility of these three movements and separation, Newton's second law was employed. The forces acting on an apex seal are shown in Fig. 4. The value of $F_s$ was chosen by measuring the apex seal deflection at 25 characteristic locations within the rotor housing and then measuring the amount of force required to deflect the seal spring for the given deflections. $F_s$ was determined to be 5.5 lb, and nearly constant over the entire operating range. This nearly constant force is accounted for by the small amount of radial seal movement over the cycle and the value of the apex seal spring constant. The force ($F_N$) used against the side seal was 5.0 lb.

Additional forces arising from rotor motion due to changes in bearing clearance or thermal distortion of the rotor housing are beyond the scope of this work and have not been included. In the analysis, the problem has been assumed to be two-dimensional with the forces $F_c$, $F_n$, and $F_s$ being resultant quantities. Their frictional counterparts $wFC$, $wFN$, and $wFS$ were determined using the assumption of Coulomb friction. The pressures $P_a$ and $P_b$, are those acting in the leading and trailing cells respectively. The values of $P_a$ and $P_b$ are input to the analysis from the experimental data. They are assumed to be uniform throughout the working cell including the regions next to the seal. Summation of the forces in the $x$ and $y$ directions yields:

$$F_c = F_N + F_s \sin \varphi - wFC \cos \varphi$$

$$+ P_a[A \cos B - A \cos \varphi - h]$$

$$+ P_b[A \cos \varphi - A \cos \varphi]$$

$$F_n = F_s - F_c \cos \varphi - wFC \sin \varphi + wFN$$

$$+ P_a[B - A \sin \varphi] + P_b[A \sin \varphi - B]$$

where $2a$ is the angle subtended by the cylindrical seal head and is equal to $\sin^{-1} (B/2A)$, $w$ is the width of the rotor, while $\varphi$ is the angle of seal obliquity and is given in Ansdale as:

$$\varphi = \cos^{-1} \left( \frac{R + 3e \cos 2a}{\sqrt{9e^2 + R^2 + 6eR \cos 2a}} \right)^{1/2}$$

In the above equation $R$ and $e$ define an epitrochoidal cylinder with $2(R + e)$ as the major diameter and $2(R - e)$ the minor diameter. A rotary engine's (Wankel) above has a trochoidal shape. A trochoid may be generated from an epitrochoidal by maintenance of a constant perpendicular distance between the two. In the case of a rotary engine this distance corresponds to the seal head radius $A$. The major diameter of a Wankel engine is the term $2(R + e + A)$ and the minor diameter $2(R - e + A)$. In Eqs. (1) and (2), seal contact between the seal and its retaining groove was assumed to be perfect. This assumption allows gas penetration into the contact region and assumes zero pressure acting on the contacted portion of the seal.

Knoll, Vilmann, Schock, and Stumpf

2
The \( \xi \) and \( \eta \) coordinates were assumed to be always parallel and perpendicular to the seal's sides, respectively.

The acceleration of the seal's cg, which was assumed to lie at one half the seal's height, was determined by direct differentiation of the equations describing the epitrochoidal path which the cg follows. These equations are derived from the equation describing the trochoidal bore:

\[
\begin{align*}
\xi &= \xi_0 + \xi_1 \cos \alpha + \xi_2 \sin \alpha \\
\eta &= \eta_1 \sin \alpha + \eta_2 \cos \alpha
\end{align*}
\]

Differentiation of these equations twice results in:

\[
\begin{align*}
\ddot{\xi} &= \ddot{\xi}_1 \cos \alpha - \dot{\xi}_1 \sin \alpha \\
\ddot{\eta} &= \ddot{\eta}_1 \sin \alpha + \dot{\eta}_1 \cos \alpha
\end{align*}
\]

which represents the acceleration of the seal's cg when the engine runs at a constant speed \( \omega \). These accelerations are related to the fixed \( x-y \) coordinate frame shown in Fig. 4. In applying the linear form of Newton's second law, it was convenient to use the \( \xi \) and \( \eta \) directions. Thus these acceleration components were transformed using:

\[
\begin{align*}
\ddot{\xi}_c &= \ddot{\xi}_1 \cos \alpha + \ddot{\eta}_1 \sin \alpha \\
\ddot{\eta}_c &= -\ddot{\xi}_1 \sin \alpha + \ddot{\eta}_1 \cos \alpha
\end{align*}
\]

If we denote the resultant of the pressure loading on the seal as \( P_c \) and \( P_n \) defined as:

\[
P_c = P_b \xi_1 \sin \alpha - A \cos \alpha
\]

\[
P_n = (P_a - P_b) \eta_1 (A \cos \alpha - h)
\]

then Newton's laws yield:

\[
F_c = \frac{m(\dot{\xi}_0 - \dot{\xi}_1) + \eta \dot{\eta}_1 + \dot{\xi}_1 P_c + F_S}{\xi_1 \sin \alpha + (1 - \eta \dot{\eta}_1) \cos \alpha}
\]

\[
F_N = m \ddot{\xi}_c + \ddot{\xi}_1 P_c + F_c (\cos \alpha - \sin \alpha)
\]

where \( m \) is the mass of the seal. In that portion of the cycle where the seal is in contact with the bore, \( F_c \) positive, there is no relative motion between the seal and the groove in the \( \xi \) direction and \( \eta \) equal to zero. Setting \( \eta \) equal to zero while the seal is in contact with the bore neglects sticktion which allows for a frictional force on the seals side with a magnitude lower than \( \eta \eta_F_c \). In a dynamic situation when small vibrations are present it was felt that this sticktion force would quickly relax allowing the seal to assume a minimum energy state. The angular form of Newton's second law was also employed to find the location of \( F_N \):

\[
\ddot{\xi} = (\frac{1}{2}) \dot{\xi}_1 \xi_1 \xi_1 \sin \alpha - \eta \ddot{\xi}_1 \cos \alpha
\]

\[
\ddot{\eta} = \eta_1 \eta_1 \sin \alpha + \ddot{\eta}_1 \cos \alpha
\]

With the forces and locations of the resultants acting on the seal known, the three types of movement were examined. With the seal initially in contact with the bore, the values of \( F_c, F_N \), and \( \dot{x} \) were calculated at the successive sampling points of the pressure data. Whenever \( F_N \) changes sign or \( \dot{x} \) obtained a value less than \( -h/2 \) the seal had shifted to the opposite side of its groove. For the locations where the seal was in contact with the leading edge of the groove, the force equations were adjusted to show the changed pressure distribution. Separation initiated whenever \( F_c \) attained a negative value. From that point in the cycle until the location where the seal recontacted the bore, \( F_c \) was set to zero in Eq. (11) and \( \eta \dot{\eta}_1 \) was calculated. Integration of the time dependent \( \eta \dot{\eta}_1 \) allowed the amount of seal separation to be traced. Since pressure data was available only at distinct points, this integration was carried out numerically. The direction of the frictional force was adjusted through the numerical integration such that motion of the seal relative to its channel was always retarded. For the operating conditions studied the maximum contact force was nominally 440 Newtons (100 lbf) and occurred when the apex seal was located from 540° to 620° ATDC.

For each of the engine conditions described in Table 1, seal separation was examined. The seal dimensions corresponding to the engine tested were input into the previous equations and are listed in the appendix. For this discussion the 0° rotor reference position corresponds to the minimum chamber volume for a system of interest on the intake stroke. The angular position depicts the motion of the trailing apex seal.

Figure 5 shows a 0° crank angle position of the apex seal under discussion and the position of this seal at various crank angle locations. From
analysis of the motoring data, it was found that lateral movement of the apex seal from the leading to the trailing side of its channel occurred at approximately the same location for all runs. This switch occurred close to 468 mainshaft degrees or at a position slightly after the major axis of the engine as the seal approaches the ignition side. In this position, the cell trailing the seal is in the intake portion of its cycle, while the leading cell is undergoing compression. The analysis predicted no lift off from the bore at this location. Also in the motoring mode, the returning shift from the trailing to the leading wall was predicted to occur in the region of 844 to 861 mainshaft degrees. This shift was accompanied by seal lift off in four of the six motoring conditions tested and occurred between the minor and major axis of the bore on the ignition side. From analysis of the firing data, which was only taken at the 100 percent throttle position, it was learned that seal transfer from the leading to the trailing side of the channel occurred in the range of 528° to 545° of crank angle. This shift also occurred after the major axis of the housing, as the seal moved toward the ignition side. No seal separation from the bore accompanied this transfer. The predicted shift from the trailing to the leading side occurred in the range of 896° to 920° of crank angle. This location occurred later in the cycle as engine speed increased. Separation from the bore was predicted to be present at the lowest speed tested and disappeared as engine speed increased. A tabulation of these results is shown in Table 2. A comparison was made between the experimentally drawn conclusions of Matsuura (7, 10) and the results presented here. Matsuura reports a return shift from the trailing toward the leading side of the channel after the minor axis of the engine on the ignition side. He also reports the return shift to occur between the minor and major axis, with this movement occurring later with increasing engine speed. These observations were confirmed with the exception of change in location of the leading to the trailing shift with engine speed. Matsuura also reports that the seal may lose contact for as much as 30° of crank angle. When the seal dimensions used by Matsuura were utilized along with data which produced a long separation (2053 rpm at 66 percent throttle), 26° of lift off was predicted. One of Matsuura's conclusions which was difficult to confirm, was his observation of increased gap with increased engine speed after 2000 rpm. This analysis shows the opposite to be true. At high engine speeds, during firing conditions, no separation occurs.

FRICTIONAL LOSSES ASSOCIATED WITH SEALING

A dynamic analysis of the forces involved with the apex seal and a knowledge of the cell pressures, allows the calculation of the frictional losses associated with the sealing grid.

In this development the assumption of Coulomb friction will again be made and any end effects due to the complicated juncture of the side and apex seals will be ignored. The basic definition of work is

\[ W = \int F \, dr \]  \hspace{1cm} (14)

will be employed in the consideration of both apex and side seals.

In the case of apex seals \( F \) in the preceding equation becomes the frictional component of the contact force, \( w_c \), and \( dr \) becomes a differential displacement along the trochoidal bore when frictional losses are calculated. The trochoidal bore is defined by adding a perpendicular displacement of magnitude \( A \) to the expressions defining an epitrochoid, given in Eqs. (5) and (6):

\[ x = e \cos^3 a + R \cos a + A \cos(a + \psi) \]  \hspace{1cm} (15)

\[ x = e \sin^3 a + R \sin a + A \sin(a + \psi) \]  \hspace{1cm} (16)

The displacement \( dr \) may be defined in terms of the angular displacement \( d\alpha \) utilizing

\[ dr^2 = dx^2 + dy^2 \]  \hspace{1cm} (17)

and differentiating Eqs. (14) and (15). With the aid of Eqs. (14) to (16), the integral defining the work done by an apex seal (Eq. (13)) may now be evaluated as an integration over the angle \( \alpha \). Since \( F_c \) is known at the \( 24 \) points where the cell pressures have been sampled, this integration was performed numerically for each of the operating conditions shown in Table 1. The numerical integration was carried out utilizing a sixth order Simpson's rule.

Frictional work associated with the side seals was computed in a similar way. Two side-sealing configurations were examined. The first model used consists of 12 side seal segments per rotor. Outer side seals, those closest to the rotor face, were assumed to be pushed against the end housing by the combined action of a spring and the gas pressure from their cells. The inner side seals were assumed to be kept in contact with the end housings by the action of a spring alone. The second side sealing system was similar to the first except the frictional contribution of the inner seal was neglected. Since the frictional work done at the side seals varies with location on the seal, this work must be calculated by evaluation of the double integral:

\[ W = \int \int dF \, ds \]  \hspace{1cm} (18)

Here \( dF \) is a frictional force acting on an incremental length \( ds \) of the seal, and \( dr \) is the distance through which this frictional force moves.

Knoll, Vilmann, Schock, and Stumpf
The distance between the rotor face and seal location is accounted for in the model by decreasing the radius input to represent the form of the side seals. The equations for this are given by:

\[ x = e \cos \theta + (g + e) \cos \left( \frac{\theta}{2} \right) - (g - e + r) \cos \left( \frac{\theta}{2} - \alpha \right) \]

\[ y = e \sin \theta + (g + e) \sin \left( \frac{\theta}{2} \right) + (g - e + r) \sin \left( \frac{\theta}{2} - \alpha \right) \]

where

\[ g = eR \left[ 1 + \cos \left( \frac{\theta}{2} \right) \right] \left[ 1 + \cos \left( \frac{\theta}{2} \right) \right] - 2e \]

\[-27.07^\circ < \gamma < 27.07^\circ \]

These equations account for the distance between the rotor face and seal location. The work integrals, Eqs. (13) and (18), were evaluated numerically. Simpson's rule was used to evaluate the double integral. Once the work integral was obtained, the frictional work associated with the bearings was known for one rotor revolution. It is then necessary only to multiply by engine speed to determine the frictional power. This analysis allows the double integral to be evaluated numerically. The present analysis predicts GSMEP of 62.7 kPa and 63.4 kPa for 2053 rpm and 2929 rpm, respectively. A 14 gram seal mass was used to approximate the mass of the steel seals. The pressure inputs to the model were those recorded while motoring the engine with the carbon apex seal. The similar values of the GSMEP predicted, is accounted for by the fact that pressure dominates the GSMEP calculation and the pressure traces during motoring for the two cases of interest were nearly identical. As can be seen in Figs. 6 and 7, GSMEP is only modestly affected by seal mass. It has been stated by other researchers that the gas sealing components may account for over 50 percent of the total frictional losses. Using this assumption and considering the fact that the coefficient of friction is not well known, these results would appear to be in good agreement.

CONCLUSIONS

The primary conclusion to be drawn from this work concerns the importance of having accurate values of the cell pressure near the seal tip. Previous analytical studies have not had cell data of this type and were unable to predict the seal separation which was experimentally known. It has been demonstrated that a dynamic analysis can yield reasonable results, when average cell pressures are input for the pressures acting near the seal. If the pressures acting in the vicinity of the seals were known, even more accurate predictions of separation could be made. This analysis has also shown that a rotation may initiate seal transfer between the side of the apex seal channel. Without considering rotation to initiate movement, it was found that under most operating conditions tested, seal transfer from side to side was not predicted. Further, these calculations show the frictional losses due to the side seals are on the same order of magnitude and possibly larger than apex seal frictional losses. Finally, it has been shown that a dynamic analysis of the sealing system can result in reasonable predictions of the

Knoll, Vilmann, Schock, and Stumpf
frictional losses associated with the gas sealing system.

REFERENCES


Knoll, Vilmann, Schock, and Stumpf
TABLE 1. - CELL PRESSURES AND OPERATING CONDITIONS TESTED

<table>
<thead>
<tr>
<th>Engine speed</th>
<th>Operating condition</th>
<th>Throttle position, percent</th>
<th>Maximum cell pressure, kPa</th>
<th>Minimum cell pressure, kPa</th>
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<td>2053</td>
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TABLE 2. - APEX SEAL MOVEMENT

<table>
<thead>
<tr>
<th>rpm</th>
<th>Throttle, percent</th>
<th>Leading to trailing shift, deg</th>
<th>Separation; distance</th>
<th>Trailing to leading shift, deg</th>
<th>Separation; distance</th>
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*M - Motoring condition.

**F - Firing condition.
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<th>rpm</th>
<th>Throttle, percent</th>
<th>(2 side seals) GSMEP, kPa*</th>
<th>(1 side seal) GSMEP, kPa**</th>
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<td>126.3</td>
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*μ = 0.1 side seals.

**μ = 0.05 apex seal.
Figure 1. Typical sealing grid for a rotary combustion engine.

Figure 2. Transducer locations.
Figure 3. - Typical working cell pressure trace.

Figure 4. - Forces acting on an apex seal.
Figure 5. - Trochoidal characteristic dimensions and labeling.

Figure 6. - GSMEP for 2053 rpm motoring.
Figure 7. - GSMEP for 2939 rpm motoring.

Figure 8. - GSMEP for Firing at 100 percent Throttle.
APPENDIX

APEX SEAL DIMENSIONS
B = 2.92 mm
h = 8.38 mm
w = 69.85 mm
A = 1.52 mm
m = 7 g

ROTOR DIMENSIONS
e = 15 mm
R = 103 mm

Figure 9. - Apex seal characteristic dimensions.