**“Reduced Engine Friction and Wear”
Semi-Annual Technical Progress Report
Second 6 Months: April 1, 2002 - September 31, 2002**

**Ron Matthews, PI, Prof. of Mech. Engineering
July 8, 2003**

DE-FG26-01NT41337

**The University of Texas
1 Longhorn Station, C2200
Austin, TX 78712**

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

ABSTRACT

Progress was made on experimental and numerical subtasks during the second 6-month period of this project.

One of the experimental subtasks scheduled for completion during the first 6 months was delayed due to a delay in receiving the prototype RLE face seal from the vendor. This component was acquired and testing was completed during the second 6 months. The test results indicate that this face seal fulfills the engineering objectives. The other experimental subtask scheduled for completion during the second 6-month period was final assembly of the prototype rotating liner engine. This subtask was completed on schedule. No results from this subtask were scheduled for this reporting period.

One numerical subtask, development of the governing equations, was scheduled for completion during the first 6-month period but was completed during the second 6 months. However, we expect to re-explore these as we learn more throughout the course of the project. Two other numerical subtasks were scheduled to begin during the second 6 months: formulating the numerical equations governing piston assembly friction and coding/testing the resulting model. These subtasks were not scheduled for completion during this reporting period. Satisfactory progress was made.

TABLE OF CONTENTS

Abstract	iii
Table of Contents	iv
List of Figures	v
1. INTRODUCTION	1
2. EXECUTIVE SUMMARY	2
3. EXPERIMENTAL AND NUMERICAL TASKS	3
3.A. <u>Experimental Tasks</u>	3
3.B. <u>Numerical Tasks</u>	7
4. RESULTS AND DISCUSSION	8
5. CONCLUSIONS	17
REFERENCES	18

LIST OF FIGURES

Figure 1. Tasks and timelines, highlighting the first year of the project	1
Figure 2. Layout of the face seal installation in the enclosure of the test rig	5
Figure 3. Overall layout of the face seal experimental test rig	6
Figure 4. RLE face seal, rotating liner, and liner drive mechanism in the prototype engine	7
Figure 5. Predicted time required for the gas/oil interface to move from the inner to the outer circumference of the sealing zone as a function of gas pressure	8
Figure 6a. Predicted face seal friction torque as a function of rotational speed	9
Figure 6b. Estimated face seal friction power and FMEP as a function of rotational speed	10
Figure 7. Comparison of the predicted and measured pressure histories	11
Figure 8. Pressure history with heated oil and both higher pressurization and higher speed	13
Figure 9. Tilt around the circumference of the RLE face seal	15
Figure 10. Measured friction torque compared to model predictions	15

1. INTRODUCTION

The US Department of Energy established the Advanced Natural Gas Reciprocating Engine (ANGRE) program to improve large-bore stationary natural gas engines. The goals of the ANGRE program are to increase fuel efficiency to 50%, decrease emissions of the oxides of nitrogen (NO_x) by a factor of 10, and decrease maintenance costs by 10. Very little work has been done to decrease the frictional losses of medium speed heavy-duty engines. Decreasing these losses is essential to improving efficiency and decreasing fuel consumption. Engine friction is dominated by the piston assembly (rings, skirt, and liner). Lawrence (1988) reports that 1% of the total energy consumption in the U.S. is due to piston ring friction alone. Ring/liner friction is also the major source of engine wear. Thus, decreasing piston assembly friction will benefit both fuel consumption and durability.

Via the ANGRE program, DOE funded a project at The University of Texas (UT) that consists of two simultaneous tasks: development of a new model for piston assembly friction and application of this model to an experimental study of a new concept for decreasing piston assembly friction. The new concept is the rotating liner engine (RLE).

Figure 1. Tasks and timelines. The vertical dash-dot line highlights the first year of the project.

The tasks and timelines that were incorporated in the proposal for this project are provided in Figure 1. The vertical dash-dot line highlights the first year of the project. The beginning of each task is an open circle and the end is a filled circle. Two subtasks were scheduled to begin during the second 6 months (Subtasks 2.2 and 2.3). Also, one subtask from the first 6 months was delayed (Subtask 1.1), but was completed during the second 6 months, and Subtask 2.1 was scheduled for completion during the second 6 months but was delayed. Subtask 1.2 was completed on schedule during the second 6 months. Progress on each of the tasks is discussed in Section 3.

2. EXECUTIVE SUMMARY

Progress was made on experimental and numerical subtasks during the second 6-month period of this project.

.Only one subtask was scheduled to produce results during the second 6-month period. Specifically, the design of the RLE face seal was to be finalized. This component seals the interface between the rotating cylinder liner and the stationary head, and is the major engineering challenge to the rotating liner concept. This seal must have low gas leakage out of the combustion chamber, low oil leakage into the combustion chamber, and low friction. The prototype seal was tested using a benchtop apparatus. These experiments showed that the RLE face seal has low gas leakage and low friction. It was not possible to directly prove low oil consumption using this experimental system, but the results indicate that this seal also fulfills this requirement. Tests in the prototype rotating liner engine are scheduled for the third 6-month period.

The other experimental subtask that was scheduled for completion during the second 6-month period was final assembly of the prototype rotating liner engine. This subtask was completed on schedule. No results from this subtask were scheduled for this reporting period.

One numerical subtask, development of the governing equations, was scheduled for completion during the first 6-month period but was completed during the second 6 months. However, we expect to re-explore these as we learn more throughout the course of the project. Two other numerical subtasks were scheduled to begin during the second 6 months: formulating the numerical equations governing piston assembly friction and coding/testing the resulting model. These subtasks were not scheduled for completion during this reporting period. Satisfactory progress was made.

3. EXPERIMENTAL AND NUMERICAL TASKS

Progress on each of the experimental and numerical tasks that were scheduled for the second 6 months of this project is discussed in the following two subsections.

3.A. Experimental Tasks

Two experimental tasks were completed during the second 6 months of the project. Each is discussed below.

Subtask 1.1, was to have been completed during the first 6 months, was delayed due to a delay in receiving prototype parts from a vendor. When the prototype face seal finally arrived, it was tested in a benchtop apparatus. A benchtop apparatus was used, rather than trying to validate the design of the face seal in an engine, for several reasons. Most importantly, it allows us to explore designs of various face seals without the need to tear down and reassemble the engine each time a new face seal design is developed. Figure 2 shows the detail of the face seal ring installation in the seal test rig. The two end retaining plates are 25.4 mm (1 ") thick steel plates and are held together by four 25.4 mm (1 ") diameter cylindrical posts (not shown in the drawing, since the drawing is a cross section at the middle). The 101 mm (4 ") long cylinders have 19 mm (0.75 ") threaded extensions, and nuts with washers hold the two plates together and contain the pressure forces. The rear flange is driven by the shaft at the back and simulates the top flange of the rotating liner. The inside of this flange is also the chamber that holds gas pressure. The rotating flange is made out of steel, but its face is composed of aluminum-bronze. A 12 mm (0.5 ") thick ring of aluminum-bronze is attached to the steel end plate. The flange is supported by an angular contact thrust bearing (SFK 7307) in order to allow free rotation. The stationary flange represents the cylinder head of the RLE engine and has all necessary O-ring glands and O-rings for the face sealing ring installation. The face seal ring and the two O-ring glands shown in Figure 2, but the O-rings are not. Oil is pumped into the space to the right of the face seal ring (the passage that feeds oil is not shown) just like in the engine, and small coil springs are installed between the flange and the face seal ring to increase the pre-load. The oil pressure is achieved via an electric oil pump with a built in pressure regulator and filter. The oil pressure is monitored by a pressure gage (not shown). The stationary flange is supported by another similar thrust bearing in order to allow for small rotary displacement. Finally, the distance between the two flanges (that determines the important dimension of upper O-ring gland to seal face) is adjusted by a spacer between the rotating flange and the rear thrust bearing.

As shown in Figure 3, the friction torque is measured via the reaction force that is applied on a moment arm attached to the flange. This is the force that prevents the rotation of the enclosure, and thus equals the friction torque that is applied to the enclosure through the face seal. Compressed air is inserted from the front of the stand, through the middle of the stationary flange. An electronic pressure transducer (PX181-300G5V, accuracy within +/-7 kPa or 1 psi) used in conjunction with a mechanical gage is used to measure and record the gas pressure history in the chamber, while the control valve (SWAGELOK BS-61-MH) allows regulation of the pressure in the chamber. A shutoff valve is also installed for fast interruption of the flow of intake air.

Figure 2. Layout of the face seal installation in the enclosure of the test rig.

Figure 3 shows the overall layout of the experimental setup. A DC electric motor (Prestolite MBD 5113) with an appropriate controller (Widdle Evolutions Inc. Model GTK 2004) energizes the setup via a pulley reduction gear. A 4:1 reduction via the pulleys is necessary in order to achieve good speed control at the very low speeds the experiment requires. A 60-tooth gear and a magnetic sensor (not shown) are used to measure the rotational speed. The moment arm that measures the reaction torque on the stationary flange is also shown in Figure 3. A spring scale is used to measure the force, while the moment arm can be readily adjusted for measuring different magnitudes of torque. The moment arm causes an overall imbalance, which is compensated for. All of the test rig components are attached to a 12 mm (0.5 ") steel plate. Adjustment for the belt tension is provided. The oil pumped into the stationary flange will readily flow through the seal and be expelled by the rotating flange. Three transparent plastic plates are attached on the 3 open areas of the enclosure assembly to confine the expelled oil. The oil flows out of the setup into a large shallow pan underneath. This pan has a hole drilled, allowing the oil to flow through the hole and into a container underneath the table. This container also acts as the reservoir for the oil, allowing oil circulation.

Figure 3. Overall layout of the face seal experimental test rig. The enclosure is shown in more detail in Figure 2.

Subtask 1.2, construction of a prototype rotating liner engine, was also completed during the second 6 months. A photo of the engine is provided as Figure 4. The engine is a 4-valve, 4-cylinder GM engine. It was converted to a single cylinder using bob weights replace the reciprocating mass of the pistons and connecting rods that were removed. This is common practice for engine research. The engine has also been converted to operation on compressed natural gas. Tests are scheduled to begin during the third 6-month period.

Figure 4 shows several features of the prototype rotating liner engine (RLE). The RLE face seal is shown as installed in the cylinder head. Also, the liner is shown in the block. The top flange of the rotating liner incorporates a gear on its periphery. This gear is driven by another gear on a shaft that projects through the spark plug hole of the adjacent cylinder. This shaft is driven by an electric motor so that the effects of liner speed on friction can be determined. Additionally, there is a torque cell on this drive shaft to allow quantification of the friction torque required to rotate the liner. In practice, once the optimum liner speed is known, the rotating liner would probably be belt driven off the crankshaft.

Figure 4. Photo of the RLE face seal, rotating liner, and liner drive mechanism installed in the prototype engine.

3.B. Numerical Tasks

The numerical tasks during the second 6 months included formulating the governing equations (Subtask 2.1) and numerical methods (Subtask 2.2) and coding these equations and methods (Subtask 2.3). We completed development of the equations, but expect to re-explore these as we learn more throughout the course of the project (and thus the arrow for Subtask 2.1 in Figure 1). We also began developing the numerical methods and coding them. Subtasks 2.2 and 2.3 proceeded on a satisfactory pace, and were not scheduled for completion during the second 6 months.

4. RESULTS AND DISCUSSION

The only subtask that was scheduled to generate results during the second 6-month period was finalization of the face seal (Subtask 1.1).

The face seal consists of 2 “zones”: a load support zone and a sealing zone. These 2 zones are separated by a circumferential groove. Oil is supplied to this groove from the top (cylinder head) side of the seal. The supply oil is then transported to both zones. The sealing zone is on the combustion chamber side of the face seal. When the oil pressure (~515 kPa or 60 psig) is higher than the gas pressure in the cylinder, the oil flows toward the combustion chamber. If this flow continues too long, oil will exit into the combustion chamber, resulting in oil consumption. When the gas pressure is higher than the oil pressure, the gas pushes the oil back toward the groove. If this continues too long, the gas will blow out the oil from the sealing zone, providing a path for gas loss. Obviously, the goal is to maintain the gas/oil interface within the sealing zone at all times.

Figure 5. Predicted time required for the gas/oil interface to move from the inner to the outer circumference of the sealing zone as a function of gas pressure.

Figure 5 shows the predictions of a model that we developed to simulate the motion of the gas/oil interface. This figure illustrates how long it takes to blow off the oil from the sealing zone as a function of gas pressure. The predictions were made for constant gas pressure because this is how the face seal experiments are performed. It is predicted that it takes ~0.6 seconds for the oil to move completely across the sealing zone at 760 kPa (110 psig). Once the oil is blown off the sealing zone, the gas pressure should begin to decrease rapidly. Experiments were performed to confirm these predictions.

Figure 6a shows the predicted face seal friction as a function of rotational speed for two different gas pressures with 10w30 oil at 25 C. The model predicts that the seal operates within the hydrodynamic lubrication regime. Even though the friction is completely of a viscous nature, the torque is not perfectly linear with speed because the film thickness increases with increasing speed. When the chamber is not pressurized, the friction is somewhat higher than when the chamber is at 690 kPa (85 psig). This is because, within the hydrodynamic lubrication regime, friction increases roughly linearly as the pressure pushing the two surfaces together decreases. Experiments were also performed to validate the friction predictions for the final face seal design.

Figure 6a. Predicted face seal friction torque as a function of rotational speed, 10w30 oil at room temperature.

Figure 6b. Estimated face seal friction power and mean effective pressure as a function of rotational speed, 10w30 oil at 90 C, gas at 565 kPa.

Figure 6b shows the corresponding friction power and friction mean effective pressure (for the prototype single cylinder engine) but with the oil at 90 C. The power required to rotate the liner is predicted to be quite small.

Figure 7. Comparison of the predicted and measured pressure histories (20 rpm, 515 kPa or 60 psig 10w30 oil at 25 C, chamber pressurized to 821 kPa or 104 psig).

Figure 7 is an example comparison between the model predictions and the experimental data from the test rig. The results of several models are shown in this figure. The initial and final periods of almost constant pressure result from a face seal code that was specially modified for the oil/gas interface phenomena of the RLE face seal. Upon pressurization to this specific gas pressure (820 kPa, 104 psig), the RLE seal code predicts that the gas pressure remains essentially constant for ~0.5 seconds (see Figure 5), at which time the oil is blown out of the sealing zone. During this period (the time period between 3.5-4.0 seconds in Figure 7), the slight pressure drop that is predicted results from heat loss, as explained below. The RLE face seal code also predicts that the oil re-enters the sealing zone when the pressure falls to 570 kPa (68 psig). The oil begins to re-enter the sealing zone when the oil pressure (515 kPa or 60 psig) is less than the gas pressure (570 kPa) because the face seal includes a spiral groove that pumps the oil against a somewhat higher gas pressure. A second model was used to predict the temperature at the end of the ideally instantaneous pressurization process. Isentropic compression of air was assumed, yielding:

$$T_2 = T_1(P_2 / P_1)^{(k-1)/k} \quad (1)$$

where the effect of temperature on the ratio of specific heats, k , was taken into account. For the specific case illustrated in Figure 7, Equation 1 yields a peak temperature of ~530 K. After the delay period, beginning when the oil is predicted to be blown out of the sealing zone, the rate of pressure loss was predicted from the equation for Pouseille flow (e.g., Alexandrou, 2001):

$$\dot{m} = -\frac{\rho H^3}{12\mu} \frac{P_u - P_d}{L} W \quad (2)$$

where H is the height of the channel, ρ and μ are the instantaneous density and dynamic viscosity of the air (where the effect of the instantaneous temperature has been accounted for), L is the length of the sealing zone (2 mm), W is the width of the channel ($2\pi r$, where r is the inside radius of the seal relative to the center of the chamber), P_u is the instantaneous chamber pressure, and P_d is the pressure at the downstream end of the sealing zone. Here, it should be noted that (1) the Reynolds number is less than 100 for all cases considered (about 5 for this specific case) so the flow is laminar, (2) the Mach No. is less than 0.1 for all cases so compressibility effects can be ignored, and (3) the length/height ratio of the channel is of the order of 1000 while the entrance length for flow development is less than 5 times the height for all cases, so entrance length effects can be ignored. However, the initial chamber pressure is greater than twice the ambient pressure. Therefore, a shock must exist, but it must exist at the exit from the sealing zone. Thus, the downstream pressure, P_d , is the pressure just upstream of this shock, or about 2 bar. When the channel height, H , is assumed to be the oil film thickness before the oil was blown out (2.3 μm for this case), the Pouseille model predicts that the pressure will decrease at an average rate of about 14 kPa/s until it becomes low enough (~570 kPa or 68 psig) that the oil from the groove begins to re-enter the sealing zone. The air temperature during this mass loss was predicted using Equation 1. That is heat loss was not considered in combination with mass loss, but this would decrease the pressure at each instant of time. As shown in Figure 7, the predicted rate of pressure loss is too great compared to the experimental data. If it is assumed that oil adheres to the surfaces within the channel, constraining the air flow to half the gap ($H/2$), the predicted instantaneous pressure is higher than the data. Yet another model was used to explore another possible explanation for the experimental trend. In this case, it was assumed that the oil was never blown out of the sealing zone and that the pressure loss is due solely to heat loss from the high temperature air to the chamber surfaces. Obtaining the heat transfer coefficient to use for these simulations required some thought. A swirling flow is imposed on the air due to the rotation of one of the two flanges. Also, some turbulence is introduced during the rapid pressurization process. It was decided to use the Woschni heat transfer correlation for piston engines (Heywood, 1988):

$$h \text{ [W / m}^2 \text{ - K]} = 3.26 \cdot B^{-0.2} P^{0.8} T^{-0.55} w^{0.8} \quad (3)$$

where B is the diameter of the chamber (0.0922 m), P is the instantaneous gas pressure in kPa, T is the instantaneous gas temperature in K, and w is an effective gas velocity in m/s, as given by:

$$w = 2.28(2SN) + 0.308 \frac{BN_s}{2} \frac{2\pi}{60} \quad (4)$$

where S is the piston stroke (0 for the test rig), N_s is the swirl speed (20 rpm for the case illustrated in Figure 7), and $2\pi/60$ is a units conversion. Given the convection heat transfer coefficient from Equation 3, the instantaneous rate of heat loss was calculated from:

$$\dot{Q} = hA(T_g - T_w) \quad (5)$$

where A is the surface area of the chamber ($\pi BH + 2(\pi B^2/4)$), T_g is the instantaneous gas temperature (from the previous time step), and T_w is the instantaneous wall temperature. Given the convection heat transfer coefficient and assuming conduction into a semi-infinite slab of steel, it was found that the wall temperature did not vary appreciably from its initial temperature 298 K. Given the predicted instantaneous rate of heat loss, the First Law was used to calculate the instantaneous gas temperature at the current time step, and then the Ideal Gas Equation was used to calculate the instantaneous pressure at the current time step. As

shown in Figure 7, this also over-predicts the rate of decrease of gas pressure. Obviously, assuming no gas loss and no heat loss yields a constant pressure.

Therefore, the experimental data in Figure 7 can be explained either via the seal remaining intact with a low rate of heat loss or by blowing out the seal followed by a slow rate of mass loss (plus a low rate of heat loss). Additional experiments and model calculations were performed to identify which is the appropriate explanation.

Figure 8. Pressure history with heated oil and both higher pressurization and higher speed (300 rpm).

Figure 8 is another comparison, but at a higher rotational speed (300 rpm) and higher pressurization (1411 kPa absolute or 190 psig) and oil heated to 60 C. In this case, the RLE seal code predicts a film thickness of 4.57 μm and that the oil is blown out of the sealing zone almost instantaneously. However, as shown in Figure 8, the leakage rate is even lower than for the conditions in Figure 7 (note the different scales). In fact, the rate of pressure loss is extremely small for this case. Assuming either that mass loss occurs through the entire seal gap with no heat loss or no mass loss but with the “maximum” heat loss predicted via Equations 3-5 yields a predicted rate of pressure loss that is far too high. Even assuming that oil occupies half the gap yields a rate of pressure loss that is far greater than the data. Assuming that mass loss occurs through only 1/10th the gap (0.46 μm) with no heat loss still yields a rate of pressure loss that is too high, and adding heat loss to this model would make the prediction worse. Therefore, the higher speed, higher pressure data can only be explained via a low rate of heat loss, possibly in conjunction with a very low mass loss rate. That is, the results shown in Figure 8 provide confidence that the RLE face seal fulfills its gas sealing function.

It must be noted that the RLE seal code predicts that the oil is blown out of the sealing zone almost instantaneously for the conditions of Figure 8 and within ~ 0.5 seconds for the conditions of Figure 7, but the data indicate that this prediction is not correct. The model necessarily assumes a perfectly axi-symmetric and flat sealing surface. However, as shown in Figure 9, the seal was not perfectly flat – a notable tilt was measured. Tilt is defined as the relative axial distance of the outside edge relative to the inside edge. It is believed that such imperfections from fabrication of the seal, such as this “tilt wave”, may contribute to it performing better than expected.

Figure 9. Tilt around the circumference of the RLE face seal for 8 evenly spaced angular locations.

Figure 10. Measured friction torque compared to model predictions for 10w30 oil at room temperature and a chamber pressure of 584 kPa.

Figure 10 compares the predicted and measured friction for 10w30 oil at room temperature and a chamber pressure of 584 kPa absolute (70 psig). The measured friction of the RLE face seal is within the estimated experimental uncertainty of the predictions. Therefore, the results shown in Figure 10 provide confidence that the RLE face seal fulfills its function of sealing with hydrodynamic lubrication (low friction).

It was not possible to directly measure the oil sealing effectiveness of the RLE face seal. However, it was noted that very little oil (< 2 cc) was in the gas chamber after performing experiments for over 10 hours, and this much or most of this oil may have been introduced during disassembly. Further, much of the time accumulated was without pressurizing the gas chamber, and this maximizes the tendency for oil to flow through the seal and into the chamber.

5. CONCLUSIONS

Only one subtask was scheduled to produce results during the second 6-month period. Specifically, the design of the RLE face seal was to be finalized. This component seals the interface between the rotating cylinder liner and the stationary head, and is the major engineering challenge to the rotating liner concept. This seal must have low gas leakage out of the combustion chamber, low oil leakage into the combustion chamber, and low friction. The prototype seal was tested using a benchtop apparatus. These experiments showed that the RLE face seal has low gas leakage and low friction. It was not possible to directly prove low oil consumption using this experimental system, but the results indicate that this seal also fulfills this requirement. Tests in the prototype rotating liner engine are scheduled for the third 6-month period.

REFERENCES

- Alexandrou, A.(2001), *Principles of Fluid Mechanics*, Prentice Hall, Up Saddle River, New Jersey, pp. 237-242.
- Heywood, J.B. (1988), *Internal Combustion Engine Fundamentals*, McGraw Hill, New York, pp. 679-680.
- Lawrence, J.B. (1988), ‘Effect of Cylinder Distortions and Piston Ring Design on Oil Consumption and Friction Losses in Automobile Engines’, DE-AC02-90236.