**Wear Reduction Systems
Liquid Piston Ring**

Final Report

**R.J. Raymond
T-N. Chen
L. DiNanno**

September 1990

Work Performed Under Contract No.: DE-AC21-88MC25035

**For
U.S. Department of Energy
Office of Fossil Energy
Morgantown Energy Technology Center
Morgantown, West Virginia**

**By
Tecogen, Inc.
Waltham, Massachusetts**

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1. SUMMARY

The overall objective of the program was to demonstrate the technical feasibility of achieving an acceptable wear rate for the cylinder liner, piston, and piston rings in a coal/water-slurry-fueled engine that utilized the concept of a liquid piston ring above the conventional piston rings and to identify technical barriers and required research and development.

The study included analytical modeling of the system, a bench study of the fluid motion in the liquid piston ring, and a single-cylinder test rig for wear comparison. A system analysis made on the different variations of the liquid supply system showed the desirability of the once-through version from the standpoint of system simplicity. The dynamics of the liquid ring were modeled to determine the important design parameters that influence the pressure fluctuation in the liquid ring during a complete engine cycle and the integrity of the liquid ring. This analysis indicated the importance of controlling heat transfer to the liquid ring through piston and liner to avoid boiling the liquid. A conceptual piston design for minimizing heat transfer is presented in this report.

A bench simulator was built with a flow visualization system to study the fluid motion in the labyrinth area of the liquid piston ring when it is moved with the piston in reciprocating motion. A 90° saw-tooth labyrinth was chosen based on the flow study to provide high shear rate at the wall and constant agitation in the channel to prevent solid accumulation.

A single-cylinder test rig was constructed using a commercial diesel engine as a base. The test piston was driven by the piston of the diesel engine, which was in turn driven by a variable speed motor. A fluidized-bed solid-particle delivery system was developed for the test rig to simulate ash loading on the cylinder wall. The test rig was used to test various configurations with solid particles feeding into the cylinder so that relative wear rates of piston ring and liner could be measured. Results showed that the liquid piston ring effectively reduced the solid particles on the wall by scrubbing, especially in the case where a surfactant was added to the water. The wear rates were reduced by a factor of 2 with the liquid ring. However, leakage of the contaminated liquid ring material past the top ring limited the effectiveness of the liquid ring concept.

On the basis of this study, it was concluded that the liquid piston ring concept has significant technical merit in reducing the wear of piston ring and liner in a coal/water slurry-fuel engine. The research and development to be carried out next is to design and develop improved piston ring seals and a liquid buffer system to prevent the leakage of the liquid ring material past the piston ring.

2. INTRODUCTION AND OBJECTIVES

There are three major technical questions to address in burning a solid fuel contaminated with solid noncombustibles (ash) in a reciprocating engine. The first question concerns combustion time. Is there sufficient time to burn the fuel efficiently for the desired fuel particle size and engine speed? This question has been answered in the affirmative for slow-speed engines burning coal/water slurries where the coal particle sizes are in the 5 to 15 μm range.¹ Current DOE-sponsored work in this area with medium-speed diesel engines has also shown that medium-speed engines can burn somewhat smaller particle sizes efficiently.²

The second question relates to component wear rates. How can economically acceptable wear rates be maintained in the mechanical components of the engine that come into contact with the ash contained in the fuel?

The third question relates to the first cost of an engine developed to burn coal. What is the cost of an engine that burns coal with an acceptable efficiency and acceptable wear rates?

In answering all of these questions there are obvious trade-offs to be evaluated, such as: fuel cost vs wear rate (ash removal) and engine first cost vs wear rate (how much complexity and/or exotic materials are built into the engine to give an "acceptable" wear rate). Developing an economically viable reciprocating engine/coal fuel system is a process of examining a large number of alternatives and synthesizing a solution. The resulting engine/fuel system can then be compared to other types of prime mover/fuel systems capable of performing the same function to determine if it is a viable solution.

This report covers the control of wear rates in a reciprocating, coal-fueled engine; specifically, wear in the cylinder, piston ring, and liner area. The approach investigated here was developed on the idea that an engine designed to burn coal will resemble an oil-fueled engine in all respects not directly related to burning coal; i.e., it will have a conventional valve train, trunk-piston arrangement, etc. This idea is based on the trends found in reports detailing DOE-sponsored work that indicate that the major contractors are modifying engines from their existing product line to make them compatible with burning coal.

The concept examined here, then, is an attempt to attack the problem of cylinder liner and ring wear in a reciprocating, coal-fueled engine by providing a solution that will have minimum impact on engine design and materials of construction.

There are two approaches to dealing with the ash present in the engine cylinder after combustion. The first approach, favored by most firms currently active in the area, is to use in the construction of the engine component materials that can operate with ash present at rubbing surfaces and retain acceptable wear rates. The second approach is to prevent the ash from coming into contact with surfaces that are in relative motion, thereby allowing the use of conventional materials.

Most firms currently engaged in coal-fueled engine work are pursuing the approach of using hard materials in the rubbing components of the cylinder and piston assembly to reduce wear rates to an economically acceptable level. Alumina, silica, and iron oxide, the major components of most coal ash, are among the hardest materials known to man, and thus cause difficulties and uncertainties in this approach.

The question also arises: What happens to the ash particles once they work their way past the piston rings and liner? In the case of a trunk-type piston engine, the ash must find its way into the sump where it will mix with the lubricating oil. Once mixed with the lube oil, the ash could conceivably be removed before the oil is fed to the bearings. However, oil contaminated with ash dripping from the piston and liner would splash onto surfaces such as cams, cam followers, and gears before ever reaching the sump. What this really means is that if the approach is to allow ash to come into contact with the rubbing surfaces, not only will the piston rings and liner require ash-compatible materials but so will many, if not all, of the remaining rubbing surfaces in the engine.

The approach of preventing the ash from ever getting between rubbing surfaces in the engine is attractive because it would allow conventional materials to be used, thereby potentially reducing development time and the ultimate cost of the engine. The most direct way of achieving this would be to prevent the ash from ever depositing on the cylinder bore. This is not considered to be a practical alternative. The next way would be to remove the ash from the bore in advance of the top piston ring, which is the approach taken in the concept evaluated in this report.

The overall objective of the program described in this report was to demonstrate the technical feasibility of achieving an acceptable wear rate for the cylinder liner, piston, and piston ring in a coal/water-slurry-fueled engine utilizing the concept of a liquid piston ring above the conventional piston rings to scrub ash particles from the cylinder walls and remove them from the engine. The definition of what constitutes an acceptable wear rate will be left to the engine manufacturer responsible for the complete coal-fired engine, since the first cost of the engine,

the fuel cost, and the maintenance cost (which is related to wear rate) are interrelated. Obviously, a very cheap fuel (relative to oil) can allow higher first cost and maintenance costs; therefore, it would be unrealistic to expect piston ring life in an engine running on coal fuel to be equivalent to that of an engine operating on Number 2 diesel fuel. For example, higher liner wear is tolerated in many modern, medium-speed diesels in order to burn a less expensive fuel.

The work described in this report includes analytical modeling, proof-of-concept bench tests, and the conceptual design of a full-scale piston. Firing tests were not carried out in this phase of the program. The work scope is conceived as a series of logical steps culminating in a decision as to whether or not the concept is worthy of a single-cylinder firing engine test.

3. CONCEPT DESCRIPTION

Figure 3.1 illustrates the liquid piston ring concept, which is the subject of this report. A liquid ring is located above the top metallic piston ring to scrub ash particles from the path of the moving piston rings and piston skirt through relative motion of the labyrinth containing the liquid ring and cylinder liner. The clean liquid is supplied to a labyrinth seal area formed in the top of the piston. The clean liquid could be supplied above the top metal compression ring, as shown in Figure 3.2, and the contaminated liquid removed through the stationary liner rather than the moving piston if the piston were much longer (by approximately the stroke of the engine). The piston rings are designed to prevent the contaminated liquid from leaking past the rings.

An analysis of the probable ash loading of the cylinder walls (see Section 5) and the amount of clean liquid renewal of the liquid ring required to maintain a low ash concentration indicates the possibility of having the liquid ring discharge directly to the cylinder, where the ash and liquid would then be discharged with the rest of the exhaust products. This variation of the concept is shown in Figure 3.3. The advantage of the once-through system is that all of the cleanup is in the exhaust system and no separate liquid cleanup (or lube oil cleanup) is required. This also simplifies, and avoids problems with, the plumbing and valving.

The flow rate of liquid to the annulus is also dependent on the heat transfer from the combustion gas. The flow rate must be high enough to prevent boiling, which would cause the loss of the liquid ring in the annulus.

The design of the labyrinth area through which liquid is pumped controls the scrubbing velocity at the liner surface and also controls the inertia of the liquid annulus moving with the piston. The inertia of the liquid annulus and the clearances between the labyrinth and liner and top ring and liner control the amount of liquid leaking into the cylinder during deceleration of the piston. Leakage of liquid down the piston past the rings is controlled by a secondary seal behind the top metallic compression ring, as shown in Figure 3.1.

The relative velocity of the liquid in the annulus is not only an important variable in scrubbing ash particles off the liner wall but also controls the amount of heat transferred to the liquid from the hot liner and piston. If the liquid temperature becomes too high, boiling could occur, which would reduce the scrubbing effectiveness and might lead to excessive loss of fluid into the cylinder. Since the liquid ring operates at instantaneous cylinder pressure, boiling is suppressed during a substantial portion of the cycle. One positive aspect of the heat transferred to the liquid ring is that it should significantly reduce the cooling requirements of the piston and liner.

Figure 3.1 Liquid Piston Ring Concept Schematic
(Liquid Recirculation Through Piston)

Figure 3.2 Liquid Piston Ring Concept Schematic
(Liquid Recirculation Through Cylinder Liner)

Figure 3.3 Liquid Piston Ring Concept Schematic (Once-Through System)

The pump for circulating the liquid through the liquid piston ring annulus could operate either at instantaneous cylinder pressure or at a lower pressure controlled by a system of check valves, since the liquid needs to circulate only during the exhaust stroke.

Following is a brief summary of the potential advantages of the concept.

- Wear-inducing particles can be prevented from reaching the engine sump. In approaches where the piston, liner, and piston ring assembly are designed to handle these particles with acceptable wear, the particles still find their way to the crankcase where they must either be removed from the lube oil before they splash onto a cam surface, for example, or every bearing system in the engine must be designed to tolerate them.
- Exotic materials for ring, liner, and piston are not required.
- The top piston ring is protected from high temperature by the liquid ring.

4. PROGRAM DESCRIPTION

4.1 TECHNICAL ISSUES

The principal technical issues involved in this concept are the achievable effectiveness of removing particles from the walls; the control of leakage from the liquid piston ring annulus to both the engine cylinder and past the top piston ring to the inter-ring area; heat transfer to the liquid ring; and lubrication of the compression rings and piston skirt. A brief discussion of each of these issues follows:

- **Particle Scrubbing** – The basis for this concept is the assumption that liquid moving over the cylinder walls at high velocity will scrub ash particles from the wall. The obvious first step was to prove that this is, in fact, the case. Bench type tests were designed and performed to validate the concept and also provide design information about the effects of scrubbing velocity, particle size, and labyrinth design on the efficiency of particle removal.
- **Leakage** – Some leakage to the cylinder is acceptable, since the fuel is presumably a coal/water slurry. Leakage will be controlled by the detail design of the labyrinth, the diametral clearance and height of the top piston land, and the deceleration of the piston. The labyrinth is critical, since it controls the scrubbing action, leakage to the cylinder, and buildup of ash deposits in the labyrinth itself. Analytical techniques were required to develop a model of the behavior of the labyrinth and to develop scaling laws so that proof-of-concept bench tests need not be full size.

In the once-through version of the concept (see Figure 3.3), a small amount of liquid would actually be displaced from the liquid piston ring labyrinth into the engine cylinder. The quantity of liquid injection would be determined by the particle loading of the cylinder wall and the desired concentration of particles in the liquid ring. For this concept to work, the leakage could not be large compared to the desired amount of liquid injected per cycle or to the total throughput of water in the slurry fuel.

Leakage of the contaminated liquid past the piston ring would cause the ash particles to be trapped between the piston ring and the cylinder wall. This would defeat the purpose of cleaning the abrasive particles off the wall with the liquid ring. The leakage would also allow the particles to enter the oil sump and cause wear of the bearings and cams.

Therefore, controlling the liquid leakage past the piston rings is an important factor in liquid piston ring effectiveness.

- Heat Transfer – Heat from the liner and piston is transferred to the liquid piston ring where it is dissipated through liquid leakage into the cylinder, evaporation and loss of liquid in the ring, or cooling of the recirculated liquid. If too much of the liquid ring is lost due to evaporation, its effectiveness in scrubbing the cylinder is reduced. An analysis was necessary to see if measures would have to be taken to reduce the heat transfer path from the piston to the liquid piston ring. The liner could be cooled to a level that should preclude heat transfer to the liquid ring if necessary.

Boiling of the liquid is suppressed whenever the cylinder pressure is elevated, since the liquid ring operates at instantaneous cylinder pressure. It may be that the effective saturation temperature will correspond to something like the mean effective pressure of the engine.

- Lubrication – Lubrication of the piston ring is of concern because the cylinder surface is exposed to liquid. There are several factors that help the lubrication of the piston rings. First, the surface is alternately exposed to oil and liquid. During the compression stroke, the piston ring moves into the zone previously scrubbed with liquid while oil is supplied to the surface below the ring. When the pressure load of the ring is the highest during the expansion stroke, the ring is moving onto the well-lubricated surface. Second, since the piston ring does not scrape the surface completely free of oil during the downstroke, it is not likely that the liquid will displace the thin layer of boundary oil away from the surface. Chemical additives may be used to further enhance the adherence of oil to the cylinder surface. Third, even if liquid mixes into the oil at the boundary, it may still provide adequate lubrication for the piston ring.

The technical issues discussed above were addressed in this program, and any barriers must be resolved before the technical feasibility of the proposed liquid ring concept can be established. To accomplish this overall goal, the following steps were taken.

4.2 ANALYTICAL MODEL

The analytical modeling of the system consisted of establishing the operating conditions and requirements for the liquid ring system, describing the flow system,

and establishing the design criteria to meet the heat transfer and liquid leakage requirements. The approach taken in the modeling effort was that of minimum complexity to achieve these goals.

An analytical model was developed to gain an insight into the factors that affect the various functions of the liquid piston ring. The analysis proceeded from simple, one-dimensional, single-phase treatment of the liquid piston ring and became more complex only as required to meet the objectives. Three-dimensional unsteady, or boundary layer analyses were not considered once it was shown that a simpler means of gaining a physical understanding of the system was adequate.

Similarly, the problem of understanding heat transfer to the liquid piston ring did not begin with finite element techniques.

Emphasis was placed on principles of similitude so that the bench rig procedures and results retain meaning when applied to the full-scale piston.

The operating conditions of the system depend on the speed and the range of sizes of the engine, the power output, and the limitations in the engine design with respect to the coolant and lubrication system. The piston stroke and speed determine the acceleration and inertial force imposed on the liquid ring mass as the piston moves around the top dead center position. An analytical model of the liquid ring was developed to describe the motion and pressure variation in the liquid as a result of the inertial force generated by acceleration, the viscous drag on the liquid by the cylinder wall, and the pressure drop across each labyrinth. The object of the analysis was to establish the design criteria for minimal leakage of the liquid into the combustion space.

Heat transfer to the liquid ring is important, particularly in the once-through system. If heat transfer to the liquid ring were more than that required to evaporate the liquid at the supply rate, the liquid ring could not be maintained. An excessive rate of liquid flow, on the other hand, is not desirable as it may affect the combustion of the coal slurry and lower the cycle efficiency.

4.3 CYLINDER SCRUBBING BENCH SIMULATOR

The scrubbing action to remove the particles from the cylinder wall due to fluid motion in the liquid ring is difficult to predict analytically. The flow pattern is unsteady due to the reciprocating motion of the piston. A bench simulation model was used to study the fluid motion in the annulus between the piston and the cylinder wall as the piston moved up and down. Since the effectiveness of scrubbing the particles off the wall is related to the fluid velocity and shear rate adjacent to the wall surface, it was the object of this study to develop the optimum labyrinth geometry for maximum flow velocity and turbulence.

The "Cylinder Scrubbing Simulator" is described in Section 6. The labyrinth in the piston is scaled up in size for visual observation of the flow pattern by means of a video camera with lighting in a narrow plane at an oblique angle to the camera. The motion of the piston was selected so that the Reynolds number duplicates that of the medium-speed diesel engines. Particles were introduced onto the surface of the cylinder wall so that their removal by the liquid ring could be observed. In addition to the scrubbing action, it is also important that the local fluid motion prevents deposition of particles anywhere in the annulus. The parameters varied were:

- Liquid Ring Gap
- Total Ring Height
- Labyrinth Geometry - Shape, Tip Clearance, Pitch, Circumferential Partitions
- Reynolds Number
- Acceleration - Inertial Force

Because of the many geometrical variables, the piston was designed in a modular form to provide the flexibility to change the labyrinth geometry with minimum effort. The piston was moved by means of a crank mechanism driven with a variable-speed motor. The results of these tests provided the data to design the single-cylinder bench test device.

4.4 SINGLE-CYLINDER LIQUID PISTON RING TEST RIG

To determine the effectiveness in removing the solid particles from the wall and the wear characteristics of the liquid ring design, a single-cylinder wear rig was used. A single-cylinder engine was modified to provide easy replacement of the piston, ring, and cylinder liner, as described in Section 7. Liquid was supplied to the annulus ring through the cylinder wall by means of a timed injector. The unit could be motored over a range of speeds. This rig was used for verification of the labyrinth design and measurement of ring wear under various operating conditions. As a first step in determining the technical feasibility of the liquid ring concept the test rig was designed to operate only at low pressure and temperature conditions. The results of this test determine whether or not to proceed to the next test in a firing engine.

The labyrinth configuration resulting from the bench simulation study was incorporated into the piston design for verification of its effectiveness in scrubbing the particles off the cylinder wall. To do this, particles were fed to the cylinder and water was continuously supplied and removed from the annulus space while the piston was reciprocated at the simulated speed. The surface of the cylinder wall

and the piston were examined periodically to determine the effectiveness of the labyrinth design. The leakage of water into the combustion space was also determined.

The design of the piston rings, their effectiveness in sealing against the leakage of contaminated water into the crankcase, and their resistance to wear were evaluated. Although this test rig did not provide the temperature and pressure conditions of a real engine, the behavior of the piston rings, the lubrication of the sliding surfaces, and the wear of the ring provided the information required to establish the feasibility of the liquid piston ring design with respect to the reduction of engine component wear for a directly coal-fired diesel engine.

5. SYSTEM ANALYSIS

The system analysis consisted of studies in three areas. The first area defined the engine configuration, engine speed and output, the coal slurry characteristics, and the flow circuit for the liquid piston ring. This analysis provided the necessary information to determine the applicability of the liquid piston ring concept and its impact on the overall system of the coal-slurry-fired diesel engine. The second area of study was the dynamic behavior of the liquid piston ring as it moved with the piston in a reciprocating motion. The local flow pattern adjacent to the cylinder wall directly influenced the effectiveness of scrubbing the solid particles off the liner surface. However, the unsteady motion was not amenable to analytical modelling, and it was studied by means of flow visualization on a large-scale bench simulator. The analysis therefore treated only the gross behavior of the fluid motion, which affected the pressure variations in the liquid ring as a function of the engine crank angle. This information helped to determine the tendency of liquid leakage into the combustion space as well as past the piston rings. The third area was the heat transferred from the combustion gas via the piston and liner to the liquid ring. It was important to determine whether or not the liquid ring could be maintained without boiling and what design changes were necessary to reduce heat transfer in order to maintain the liquid ring. Various aspects of the system analysis are presented in the following sections.

5.1 RECIRCULATED LIQUID RING SYSTEM

The recirculated liquid ring concept is illustrated in Figures 3.1 and 3.2 for two different methods of introducing and discharging the liquid.

In Figure 3.1, the liquid entry and discharge are through the piston. This design has the advantage that the liquid entry and discharge points always remain the same relative to the liquid ring annulus. Therefore, by properly selecting the inlet and discharge ports, a flow pattern can be established such that the clean liquid is always near the piston ring while the contaminated liquid is near the top of the piston. The major challenge in this design is the method of piping the liquid in and out of the piston which is in constant reciprocating motion. The following approaches are considered technically feasible.

- Telescopic pipes with sliding seals.
- Flexible hoses.
- Pipe linkage with rotating seal at joints.

The telescopic pipes have been used on low-speed cross-head type engines for bringing cooling water to the piston. However, in the liquid ring application,

the return water contains particles of abrasive materials which may cause wear problems in the sliding seals. The pipe linkage arrangement with sealed joints could potentially face the same wear problem at the seals when they are exposed to the contaminated liquid over a prolonged period of time. The flexible hose system requires selection of the proper material for long life and special design of the connections. The inertia of the hose and the liquid within it contribute to the dynamic behavior of the flexible hose system under reciprocating motion.

Figure 3.2 shows an alternate approach in which the liquid is introduced and discharged through the cylinder wall. This design eliminates the need to pipe the liquid through the moving piston. In this design, the height of the liquid ring section above the piston ring is greater than the stroke of the piston, so that both the injection and discharge ports remain exposed to the liquid ring at all times. When the piston is at top-dead-center the ports are at the bottom of the liquid ring and just above the piston ring. At bottom-dead-center, the ports are near the top of the piston crown. This arrangement can also establish a flow of liquid in the axial direction away from the piston ring if the injection of the liquid is timed to take place near top-dead-center while discharge of the liquid occurs near bottom-dead-center. The added height of the piston has a significant impact on the overall dimensions of the engine as well as the static and dynamic force balance on the piston. A shorter liquid ring section is also acceptable if valves are provided to prevent the cylinder gas from discharging into the liquid system.

One advantage of the recirculated liquid ring system is the flexibility in selecting the liquid flow rate. The flow rate of liquid affects the particle concentration in the liquid ring, which in turn affects the probability of the particles being trapped between the piston ring and the cylinder liner. The liquid flow rate also determines the temperature rise of the liquid in the ring due to heat transferred from the combustion gas through the piston and the cylinder liner. High liquid flow rate can prevent the liquid from reaching the boiling point particularly if the pressure in the liquid ring drops below atmospheric pressure during the exhaust stroke of the cycle. This will be discussed later.

The recirculated liquid ring system involves a rather complex flow circuit external to the engine cylinders (see Figure 5.1). The contaminated liquid from the engine must be cleaned up with a filter or separator before the liquid is sent back to the engine. Liquid make-up is necessary as part of the liquid will be lost to the combustion space and also disposed with the solids. The pressure level of the liquid injection pump depends on the timing of the injection. For example, if continuous injection is desired, the liquid must be pumped to a pressure higher than the peak firing pressure in the cylinder. If one chooses to inject during intake and exhaust stroke when the cylinder pressure is low, then a low pressure pump

Figure 5.1 Flow Diagram for Recirculated Liquid System

may be used in conjunction with check valves in the supply line to prevent back flow. A timed valve would be operated according to the engine crank angle if a specific timed injection is desired.

The solid particles removed from the engine cylinder must be separated from the liquid and discharged from the system continuously. Since the particles could be in the submicron size range, a very efficient method of separation is required. The disposal could be accomplished by discharging the concentrated slurry into the hot exhaust gas so that the liquid is evaporated and the solids captured in the exhaust clean-up system.

In general, the recirculated liquid ring system, in spite of its advantages in controlling the flow and temperature in the ring, is a less preferred approach due to the complexity of the overall system. A simpler system with once-through liquid feed is described below.

5.2 ONCE-THROUGH LIQUID SYSTEM

The once-through liquid ring system, as shown in Figure 3.3, only has ports to admit clean liquid. The contaminated liquid is discharged directly into the cylinder space where the ash particles and liquid are then discharged with the rest of the exhaust products. The advantage of the once-through system is that all of the cleanup is in the exhaust system and no separate liquid cleanup is required (see the flow diagram in Figure 5.2). This simplifies, and avoids problems with, the plumbing and valving. The liquid inlet ports are located near the piston ring when the piston is at top dead center. Therefore, if the injection of liquid is timed near top dead center, the liquid flows away from the piston ring towards the top of the piston, thus keeping the contaminated liquid away from the piston ring area. In the case where the liquid ring height is less than the piston stroke, the timed injection valve is necessary to prevent the cylinder gas from flowing into the liquid supply system.

The once-through system is limited in the amount of liquid flow to the ring and subsequently into the combustion space. The addition of the liquid into the combustion gas has two effects. First, large amounts of liquid addition to the coal slurry fuel cause the combustion rate to slow down and result in lower combustion efficiency and overall thermodynamic cycle efficiency. In the extreme case, it may cause engine misfiring. The limit of liquid flow rate is difficult to predict, because it is a function of the liquid form, i.e., droplet size, and the time distribution of the liquid flow over the various parts of the cycle. For example, if the majority of the liquid is discharged into the cylinder during the exhaust stroke, there is little effect on combustion. The second effect is the addition of the liquid on the thermodynamic cycle efficiency. If the liquid is added to the fuel-air mixture and

Figure 5.2 Flow Diagram for Once-Through Liquid System

proceeds through the normal cycle of compression, evaporation, and expansion, it causes the flame temperature to drop, thus causing a reduction of thermal efficiency for the normal fuel-air charge. However, the added liquid also contributes to the total power output, and therefore the net result could be an increase in power and efficiency. Analysis and experiments made on inlet manifold water injection³ showed a slight improvement in fuel consumption up to a ratio of water to fuel of 1.25 for the case of stoichiometric gasoline-air mixture. Therefore, it is reasonable to assume that a liquid mass injection rate comparable to the fuel mass rate is acceptable for the once-through system.

5.3 ASH SYSTEM PARTICLE LOADING

The next question for the once-through system is what would be the magnitude of ash concentration in the liquid ring if the liquid flow rate is limited to the fuel flow rate. Let us consider the following conditions.

Fuel Properties

Dry Coal	50%
Moisture	50%
Particle Size (Average)	10 microns
Ash Content	0.53%
Lower Heating Value (Dry Coal)	13,740 Btu/lb
Lower Heating Value (Slurry)	6,750 Btu/lb

Engine Condition (GE)

Bore	9 in.
Stroke	10.5 in.
Speed	900 to 1200 rpm
Power	249 to 274 hp/cyl.
BSFC	6,500 Btu/bhp-hr

These figures yield the following estimates.

Coal Consumption	0.473 lb/bhp-hr
Water Loading	0.473 lb/bhp-hr
Ash Loading	0.00251 lb/bhp-hr

If 10 percent of the ash from the coal is assumed to contact the cylinder wall and is held by the oil film, then the concentration of ash particles in a homogeneously mixed liquid ring can be determined for a given liquid flow rate. Let clean liquid enter the ring at flow rate m_1 , contaminated liquid at particle concentration C leaves the ring, and particles deposit on the cylinder wall from the

combustion gas at a rate of m_g . To maintain a steady-state concentration of particles in the liquid ring, we have, for mass conservation,

$$m_s = C (m_g + m_l)$$

At a particle deposition rate of 0.000251 lb/bhp-hr

$$C = \frac{0.000251}{m_l + 0.000251}$$

Therefore, at a liquid flow rate comparable to the water loading from the slurry fuel, i.e., $m_l = 0.473$ lb/bhp-hr, the concentration of particles in the liquid ring is $C = 0.00053$. This is a reasonably diluted slurry to be maintained in the ring. The concept of the once-through system is therefore considered feasible, and further investigation is warranted in view of its advantages over the recirculated system.

5.4 DYNAMICS OF THE LIQUID RING

The two major technical issues addressed in the system analysis are the following:

- Dynamic behavior of the liquid ring, which determines the leakage of the liquid into the cylinder and the effect of the labyrinth geometry on that leakage.
- Heat transfer from the cylinder liner and the piston to the liquid ring, which affects evaporation and loss of liquid in the ring.

The heat transfer considerations are discussed in Section 5.5.

An analytical model to describe the dynamic behavior of the fluid ring was developed. The model includes the inertia force generated by the reciprocating motion as the liquid ring moves up and down with the piston, the viscous drag on the liquid due to its motion relative to the cylinder wall, and the pressure drop resulting from flow across the labyrinth. Although simplifying assumptions are made regarding the flow pattern and the viscous drag, the model helps to identify the important design parameters for the liquid ring.

For a simple annular ring model, as shown in Figure 5.3, the static pressure in the liquid at a depth L from the top free surface can be expressed as a function of the liquid ring configuration and the instantaneous engine operating parameters:

$$p - p_o = \frac{\mu C_v L V}{g h^2} + \frac{n k}{(\pi D C h)^2} \frac{m^2}{2 g p} + \frac{\rho L}{g} (a + g) \quad (1)$$

where:

- n = number of labyrinths per inch
- L = depth of liquid ring from top free surface
- h = liquid ring thickness
- V = piston velocity
- a = piston acceleration
- D = piston diameter
- m = mass through flow rate
- ρ = liquid density
- μ = viscosity of liquid
- C_v, C, k = correction constants for assumed velocity profiles

The first term on the right hand side of the equation represents viscous drag resulting from the relative motion between the cylinder wall and the liquid ring. C_v is a correction factor for the assumed linear velocity gradient. This term varies in magnitude and direction during a cycle. The second term is the pressure drop due to the net flow of the liquid across the labyrinths. It is assumed that the flow is constant and it is in the direction from the piston ring toward the top of the piston. The third term is the inertia of the liquid mass when it is moved up and down with the piston.

Figure 5.4 shows for a 9-in. bore x 10.5-in. stroke engine the variation of pressure differential between the top and bottom of the liquid ring in one revolution of the engine and the effects of the various design and operating parameters. When the ring thickness is larger than 0.0005 ft in this medium-speed engine the ring height and engine speed are the only two major controlling parameters on the pressure in the liquid. When the ring thickness is reduced to 0.0002 ft, as shown in Figures 5.4(c), (d), and (e), the labyrinth pitch also becomes effective in controlling the liquid pressure. The liquid flow rate affects the pressure only when it is appreciably higher than that corresponding to the water loading in the coal slurry fuel even when the ring thickness is only 0.0002 ft.

It is important to note that for a certain geometry and engine speed the pressure at the bottom of the ring drops as much as 7 psi below the cylinder

Figure 5.3 Model of Liquid Piston Ring

Figure 5.4 ($P_1 - P_0$) versus Crank Angle - Medium Speed Engine
9.0-in. Bore, 10.5-in. Stroke

pressure when the piston is near top dead center. This suggests that, at the end of exhaust and beginning of intake stroke, such a condition may result in boiling of the liquid if its temperature is high enough to exceed the equilibrium condition. Therefore, for a given engine, the liquid flow rate, the heat transfer to the liquid ring through the piston and the cylinder liner, and the liquid ring geometry determine the integrity of the liquid ring. Boiling the liquid near the end of the exhaust stroke could be desirable since the vapor generation helps to discharge the liquid and the suspended particles into the cylinder space and exhaust through the exhaust valves. Engine testing will be needed to determine whether or not boiling will help to scrub the wall and discharge the contaminated liquid. If so, the liquid ring can be designed to enhance the boiling condition.

Figure 5.5 shows the pressure variation for a typical low speed industrial stationary engine of 13-in. bore x 16-in. stroke. In this case, the inertia force is lower and therefore the pressure swing is lower than that in the medium-speed engine. Larger pressure depression, if desired to produce boiling effects, can be achieved by increasing the height of the liquid ring.

Discharge of the liquid into the cylinder space can occur due to two different mechanisms. One mechanism is that when boiling occurs in the liquid ring near the piston ring, where the pressure drops below the vapor pressure corresponding to the liquid temperatures, the formation of vapor pushes the liquid above it out of the annulus space. The second mechanism is that when there is circumferential variation in pressure due to variation in liquid ring thickness, a flow pattern could be developed due to the axial and circumferential pressure gradients such that the liquid flows out of the annulus over a portion of the circumference while, over the other portion, liquid flows axially into the annulus. The tendency for this flow pattern to occur depends on the axial pressure gradient in the liquid ring. The higher the pressure variations, the higher will be the tendency for the liquid to discharge into the cylinder space.

5.5 EFFECT OF HEAT TRANSFER

The integrity of the liquid piston ring depends on the temperature of the liquid and the pressure in the liquid near top-dead-center exhaust. The temperature of the liquid is a function of the rate of liquid flow and its initial temperature and the heat transfer from the piston and the cylinder liner. The objective of this study is not to carry out a detailed heat transfer analysis from the combustion gas to the piston and the cylinder liner and through the metal structure to the liquid ring. This study is to establish the order of magnitude of the heat transfer based on past studies in diesel engines and to determine what might be needed in terms of design modifications in order to reduce the heat transfer to the liquid ring.

Figure 5.5 $(P_1 - P_0)$ versus Crank Angle - Low Speed Engine
13-in. Bore, 16-in. Stroke

Heat transfer from the combustion gas to the piston was measured previously in diesel engines,⁴⁻⁸ and the heat flux ranged between 55,000 and 150,000 Btu/ft²hr. Heat flux to the cylinder liner was averaged at 40,000 Btu/ft²hr. If we take the example of the medium-speed engine, 9-in. bore x 10.5-in. stroke, and estimate the total heat transferred to the liner and the piston, we have:

Total Heat Transfer into the Piston = 24,300 to 66,330 Btu/hr

Total Heat Transfer into the Liner = 82,500 Btu/hr

Assuming for the once-through system a water feed rate of 118 lb/hr at 249 bhp and specific water feed rate of 0.475 lb/bhp-hr, it is seen that the total heat input required to vaporize the water, including sensible and latent heat, is comparable to the total heat transfer from the combustion gas. This means that one can design the piston to maximize or minimize the heat conduction to the liquid ring depending on whether or not boiling the liquid in the ring is beneficial to the reduction of piston ring and liner wear. Similarly, by controlling the temperature of the engine coolant in the liner jacket the net heat input to the liquid ring can be changed to control the boiling. If the pressure in the liquid ring drops to 7.5 psia at TDC exhaust, the liquid in the ring must be held below 180°F to prevent boiling. In this case, the temperature of the coolant in the engine jacket must be kept below 180°F to allow heat transfer from the liquid ring to the jacket.

Unless the system is actually tested in an operating engine (which was not within the scope of this project) it cannot be determined whether allowing the liquid to boil in the ring is desirable from the standpoint of piston ring and liner wear. The boiling action helps to discharge the liquid into the combustion space and also to create high velocity liquid movement and agitation to enhance the scrubbing of the ash particles from the liner wall. However, it is clearly not possible to boil the liquid near the piston ring and push out and empty the liquid ring on every revolution of the engine. It is easy to show that in the once-through system it takes a large number of revolutions to fill the liquid ring annulus when the feed rate is limited to the amount comparable to that in the coal-slurry. This means the liquid ring cannot be established within one cycle, and consequently the metallic piston ring will not be protected by the liquid ring at all times. Therefore, we may conclude that the control of heat flux to the liquid ring and the extent of boiling are important factors to consider when judging the system's wear reduction capabilities.

The following are some of the methods to control the heat transfer to the liquid ring.

- Design the piston such that the load is transmitted to the wrist pin through structural members instead of the cylinder forming the inside wall of the liquid ring so that the thickness of the cylinder can be reduced to cut down the longitudinal heat conduction to the liquid ring.
- Use a two-piece piston design that includes a heat dam at the corner between the piston crown and the labyrinth section of the liquid ring, as shown in Figure 5.6. Oil is injected between the two pieces for piston cooling. Heat transfer from the oil to the liquid ring can be controlled by oil temperature.
- Separate the coolant circuit for the cylinder jacket from that for the cylinder head. This would permit the control of the jacket coolant temperature and heat transfer to the liquid ring without affecting the cylinder head temperature.

In general, even though this study does not identify the heat transfer conditions needed to operate with the liquid ring system, we can conclude that the control of heat transfer to the liquid ring is technically feasible.

Figure 5.6 Two-Piece Piston Design with Thermal Barrier to the Labyrinth

6. BENCH SCRUBBING SIMULATION STUDY

6.1 LABYRINTH GEOMETRY

The liquid piston ring concept for reducing wear of the piston ring and cylinder liner is based on the premise that the ash particles deposited on the wall can be lifted by the liquid ring so that the piston ring in relative motion with the liner surface would not come in contact with the particles. The effectiveness of the liquid ring system for wear reduction is therefore dependent on how effectively the particles can be scrubbed by the motion of the liquid ring. The concept of using a labyrinth geometry was to promote the shear rate and turbulence adjacent to the surface. The bench simulation model was designed to study the fluid motion in the liquid ring annulus as the piston moves up and down. The test results were used as a guide to selection of the labyrinth geometry for testing in the single cylinder test rig.

The selection of the labyrinth geometry was based on two criteria. One was to maximize the shear rate and turbulence at the surface of the liner wall, and the other was to maintain agitation throughout the annulus at all times so that the particles would not have a chance to settle down and become agglomerated and eventually plug up the entire annulus ring. The geometries of the labyrinth selected for testing were also limited to the simple shapes that were considered practical from the standpoint of machining. Two basic geometries were selected for the simulator test - rectangular and saw-tooth grooves - as shown in Figure 6.1. For the rectangular configuration, the width to depth ratio of the groove was varied. For the saw-tooth case only a 90° saw-tooth was tested.

6.2 BENCH SCRUBBING SIMULATOR

To select the labyrinth geometry for maximum effectiveness in scrubbing the particles off the wall, a large-scale simulator was designed and built for the purpose of studying the flow patterns in the labyrinth region. The labyrinth was scaled up in size for ease of visual observation. The required piston speed to duplicate the Reynolds No. was reduced, which made the operation of the simulator and visual observation of the flow field easier.

The characteristic Reynolds No. for the flow field in the liquid ring, which is induced by the relative motion between the piston and cylinder wall, may be expressed by:

$$Re_h = \frac{\rho_w V_p h}{\mu} \text{ or } Re_a = \frac{\rho_w V_p A}{\mu}$$

Figure 6.1 Labyrinth Geometry

where ρ_w is the density of the liquid, V_p is the piston velocity, h is the clearance between the tip of the labyrinth and the cylinder wall, A is the pitch of the labyrinth teeth, and μ is the viscosity. If the labyrinth in the simulator is scaled up with the tip clearance, then for a constant Reynolds No. we have

$$V_p h = V_{ps} h_s$$

and $V_{ps} = V_p h / h_s$

where the subscripts designate the simulator. A tip clearance of approximately 0.025 in. was chosen for the simulator to allow sufficiently large passage for the polystyrene spheres used for flow visualization to move from one labyrinth tooth to the next. This clearance is roughly 10 times that in a real engine if a 0.0025 in. clearance is used.

It should be noted that the simulator did not fully simulate the transient phenomena due to the variation of the piston velocity and acceleration over the cycle. Since the flow pattern in the liquid ring varied with time, the time available for the development of the flow pattern would affect the magnitude of the velocity at a given time. To fully simulate the system, a very long stroke was required for the simulator, which was impractical. Since we were looking for qualitative information as a guide to evaluating geometries, a 4.33-in. stroke was accepted for the simulator.

The simulator is a piston with labyrinth in a plexiglass cylinder, as shown in Figure 6.2. The piston was built up with a stack of machined rings of the desired geometry. This construction allowed flexibility to change the geometry, particularly in the rectangular tooth configuration. The piston was attached to the top of the piston of a single-cylinder 4.33-in.-stroke diesel engine which was used to provide only the required reciprocating motion. The engine was driven by a variable-speed motor through an in-line speed reducer. A similar arrangement was used for the single-cylinder test rig, and a more detailed description of the drive system is given in Section 7. Figure 6.3 shows the overall setup of the simulator system.

During operation the cylinder was filled with water up to a level approximately 1/2 inch above the piston. An "O" ring was provided at the bottom ring to prevent water from leaking down. A flow visualization system was developed to provide visual observation of the flow pattern as well as video recording so that in high speed operation, the flow pattern could be viewed by playing back in slow motion.

Figure 6.2 Bench Scrubbing Simulator

Figure 6.3 Bench Scrubbing Simulator

6.3 FLOW VISUALIZATION SYSTEM

The flow visualization was accomplished by using polystyrene tracer beads in water with properly controlled lighting. The beads have a specific gravity slightly higher than water, but they followed the streamline quite well because of the relatively low water velocity of about 1 ft/sec. To facilitate viewing and to cut down reflected light, the labyrinth surface was painted with a non-reflective black paint. Lighting was provided from the side through a 1-in.-wide slit parallel to the piston axis so that only a narrow circumferential segment of the liquid ring was illuminated. A direction approximately 70 degrees from the viewing direction gave the best illumination. Because of distortion of the view due to the transparent cylinder, a square plexiglass box was built around the cylinder and filled with water. This arrangement completely eliminated the distortion when viewing from the flat side of the box. The flow visualization system is shown in Figure 6.4.

Since the piston and the liquid ring move up and down, it was not possible to view the flow pattern unless the viewer or the camera moved up and down with the piston. A relatively simple system (as shown in Figure 6.5) was designed with mirrors that permitted the camera to be stationary. The drawback of this arrangement was that the effective distance between the object and the viewer varied with time, which caused a slight variation in the sharpness of the picture due to changing focus. Nevertheless, it provided a sufficiently clear view of the flow pattern in the labyrinth for examination. The video record was played back at reduced speed for ease of viewing the transient flow development as the piston and the liquid ring reciprocated in relative motion to the stationary wall.

6.4 TEST RESULTS

Seven different labyrinth geometries were tested, including one 90° saw-tooth shape and five rectangular shapes. The depth and width of the channel were varied as shown in Figure 6.1. For each geometry the simulator was operated at different speeds up to 100 rpm. Above 100 rpm, difficulty was encountered when the water in the simulator tended to splash and be thrown out of the cylinder. A typical view of the flow pattern in the labyrinth region is shown in the timed exposure picture in Figure 6.6. Due to the unsteady nature of the flow pattern, the observation is represented in the form of a qualitative description, as shown in Figure 6.7(a) through (g).

In the saw-tooth configuration, a major vortex was developed quickly in the trailing corner of the triangular channel, while a secondary vortex was situated at the middle of the channel. Since the formation of the major vortex was in the opposite corner from the existing vortex of opposite direction, the time to start the

Figure 6.4 Flow Visualization Lighting System

Figure 6.5 Flow Visualization System for Stationary Camera

Rectangular Labyrinth

Saw-Tooth Labyrinth

Figure 6.6 Flow Tracer Picture of Bench Simulator Test

Pictures illustrate typical fluid motion at 30 rpm during the upstroke
 Existence: Engine speed at which vortex exists
 Size: Relationship between vortex size and engine speed
 Formation: Distance from BDC to when vortex becomes visible (approximate)
 Deterioration: Distance vortex remains visible into the downstroke (approximate)
 Center: Location of vortex center

(all dimensions are in inches)

(a) Configuration analyzed at 10-100 rpm

Upstroke

Vortex A:

Existence: 10-100 rpm
 Size: Decreased with increasing engine speed
 Formation: 1.5 inches (10-50 rpm)
 0.75 inches (50-100 rpm)
 Deterioration: 0.75-0.35 inches (10-60 rpm)
 less than 0.35 inches (60-100 rpm)
 Center: Point 1 (100 rpm)
 Point 2 (10 rpm)

Vortex B:

Existence: 20-50 rpm
 Size: Remained approximately the same
 Center: Point 3
 Note: Very weak vortex

Downstroke mirrors upstroke

(b) Configuration analyzed at 10-60 rpm

Upstroke

Vortex A:

Existence: 10-60 rpm
 Size: Does not change with engine speed
 Formation: 1.5-2.5 inches (10-60 rpm respectively)
 Deterioration: 1.5 inches
 Center: Point 1
 Note: Above 30 rpm vortex began small (with center at point 2) and grew to normal size approximately half way through the stroke.

Downstroke mirrors upstroke

(c) Configuration analyzed at 10-60 rpm

Upstroke

Vortex A:

Existence: 10-60 rpm
 Size: Increases to fill section with increasing engine speed
 Formation: 1-1.5 inches (10-30 rpm)
 Sometimes would not form (40-60 rpm)
 Deterioration: 0.5-1 inches (10-30 rpm)
 Sometimes did not deteriorate (40-60 rpm)
 i.e. vortex would not change direction when stroke changed direction
 Center: Point 1 (10-30 rpm)
 Point 2 (40-60 rpm)

Downstroke mirrors upstroke (10-30 rpm)

Figure 6.7 Observation of Flow Pattern in Labyrinth Channels

(d) Configuration analyzed at 10-60 rpm

UpstrokeVortex A:

Existence: 10-40 rpm
 Size: Height of vortex remained the same but width became thinner with increasing engine speed
 Formation: 0.35-1 inch (10-40 rpm respectively)
 Deterioration: 0-1 inch (10-40 rpm respectively)
 Center: Point 1 which moved closer to the cylinder wall with increasing engine speed

Vortex B:

Existence: 30-60 rpm
 Size: Became slightly smaller with increasing engine speed
 Center: Moved from point 2 towards point 1 with increasing engine speed

Downstroke mirrors upstroke

(e) Configuration analyzed at 10-60 rpm

UpstrokeVortex A:

Existence: 10-60 rpm
 Size: Stays approximately the same
 Formation: 0.75-1.5 inches
 Deterioration: 0-0.75 inches
 Center: Point 1

Vortex B:

Existence: 20-40 rpm
 Note: Very weak vortex

Downstroke mirrors upstroke

(f) Configuration analyzed at 60 rpm

UpstrokeVortex A:

Existence: 60 rpm
 Size: As shown
 Formation: 1.5 inches
 Deterioration: Less than 1 inch
 Center: Point 1

Downstroke mirrors upstroke

Figure 6.7 Observation of Flow Pattern in Labyrinth Channels (cont'd)

(g) Configuration analyzed at 35-60 rpm

UpstrokeVortex A:

Existence: 35-60 rpm
 Size: Vortex height remains the same
 the width becomes smaller with
 increasing engine speed
 Formation: 1.5 inches
 Deterioration: 0.35 inches
 Center: Point 1 which moves closer to
 the cylinder wall with increasing
 engine speed

Vortex B:

Existence: 35-60 rpm
 Size: Fill rear section of labyrinth
 Center: Point 2 moves towards the
 cylinder wall as engine speed
 increases
 Note: Vortex becomes larger as engine
 speed increases

Downstroke mirrors upstroke

Figure 6.7 Observation of Flow Pattern in Labyrinth Channels (cont'd)

vortex was short and the vortex grew in size as the opposite vortex decayed. The fluid in the channel appeared to be sufficiently agitated everywhere to prevent settlement of particles. Based on qualitative reasoning, one might conclude that the maximum shear rate occurred during the first half of the piston stroke when the fully developed vortex from the previous stroke was being decelerated.

With rectangular channels, the development of the vortex was generally slower because the size of the vortex was dictated by the channel depth and width. In the case shown in Figure 6.7(c) where the width is only slightly larger than the depth, the vortex filled the entire cavity. The vortex developed in the previous stroke had to be fully decelerated before a new vortex of opposite direction could be generated. This is why at higher engine speeds the vortex did not consistently reverse on each stroke. When the width to depth ratio was high and the depth was small, as in the case of Figure 6.7(e), the vortex formed on the trailing side and the development of the vortex was relatively fast. However, the vortex was small and it also deteriorated quickly. The major drawback of the rectangular shape was that the flow activity in the corners was low. Consequently, polystyrene beads tended to collect in the lower inside corner. This condition would cause accumulation of ash particles and eventually could have led to total plugging up of the labyrinth.

Based on the above results, it was concluded that the saw-tooth geometry can provide the high fluid shear at the cylinder wall and constant agitation in the labyrinth channels to prevent solid particles from settling. Therefore, the saw-tooth geometry was selected for testing in the single-cylinder test rig.

7. SINGLE-CYLINDER TEST RIG

7.1 DESCRIPTION OF TEST RIG

Figure 7.1 shows the single-cylinder test rig in a schematic fashion while Figure 7.2 is a photograph of this same facility. The test rig was built up around a commercially available Petter single-cylinder diesel engine of the following dimensions:

Bore - 3.4 in.

Stroke - 4.3 in.

The engine was driven by a variable-speed-drive system.

The cylinder head and valve gear were removed from the engine and the piston was modified to become a cross-head piston for the liquid ring piston and cylinder liner. This arrangement is shown in detail in Figure 7.3. It should be noted that the use of a cross-head was an expedient to allow evaluation of the performance of the liquid ring. (All of the present DOE coal-fired diesel engine contractors are following a trunk type piston approach.) A second piston for the same diesel engine was modified to carry the liquid piston ring labyrinth, and a second cast-iron cylinder liner was used as the test cylinder liner. Hence the liquid piston ring test cylinder had the same bore and stroke as the diesel engine.

7.1.1 Liquid Ring Feed System

The diesel engine injection pump was used to introduce clean fluid to the liquid ring area. The governor of the pump was disconnected to allow independent control of the quantity of liquid injected, and the system was calibrated at various speeds and rack settings. The injection pump was capable of supplying the liquid at high pressure, but for the test rig the high pressure was not needed since the cylinder pressure at the time of injection was only about 20 in. H_2O above atmospheric pressure. Liquid was supplied to the pump from a reservoir and through a standard fuel filter, as shown in Figure 7.1.

Liquid was injected at a single point near the bottom of the labyrinth with the piston at top dead center. There was a single injection every other revolution when the piston was near top dead center. Liquid was removed from a port opposite the injection port and near the top of the labyrinth, as shown in Figure 7.3.

Figure 7.1 Wear Reduction System Test Rig Schematic

Figure 7.2 Single-Cylinder Test Rig

Figure 7.3 Piston/Cylinder Arrangement for Wear Reduction Test Rig

A collection tank for the contaminated liquid/slurry mixture was connected to the liquid ring outlet on the cylinder liner, as shown in Figure 7.1. Since the pressure fluctuated more in the cylinder than in the collector, the pressure differential between the cylinder and collector near top dead center helped to discharge the liquid at the top of the liquid ring into the collector. Most, but not all, of the liquid and alumina introduced was trapped in this collector and in the large receiver plumbed to the test cylinder head.

7.1.2 Solid Particles Feed System

To determine the relative wear rate of the piston rings and cylinder liner with and without the liquid piston ring, an abrasive material was used for the single-cylinder rig test. Alumina particles with an average size of 3 microns (0.8 to 8.0 microns) were used. The particles were carried continuously by an air stream into the test cylinder. A portion of the particles was deposited on the cylinder wall and the remaining were discharged from the cylinder along with the air. The deposited particles were either mixed with the lube oil or scrubbed off the wall and discharged along with the water in the liquid ring.

Feeding of the particles into the air stream was not straightforward because of the small particle size and their poor flow characteristics. A fluidized bed system was developed for the feeding of the alumina particles, which gave a reasonably constant rate over the short test duration. The fluidized solid feed system is shown in Figure 7.4. It consisted of a 4.25-in. O.D. plexiglass cylinder for ease of observation with a diffuser plate separating the plenum chamber and the fluidized bed. The diffuser plate was drilled with a square array of 0.042-in.-diameter holes at 0.25-in. center spacing. The system operated at a pressure above atmospheric that was dictated by the rest of the flow system through the test cylinder and the exhaust. A solid feed hopper was provided at the top of the bed, which was periodically rotated manually into position in line with an opening to allow a batch of solids to drop into the bed. The airborne solids were fed through a 0.5-in. I.D. flexible tube into the test cylinder.

Initial attempts to fluidize the alumina particles in the bed were not successful. At low airflow, the bed was fixed. As the air flow-rate was increased, a portion of the bed suddenly became fluidized while the remaining area stayed fixed. The rate at which the solid particles were carried out with the air was high and uncontrollable. This phenomenon was caused partly by the low pressure drop across the diffuser plate at the low fluidization velocity and partly by the tendency of the small particles to cake up.

Figure 7.4 Particle Feed Apparatus for Wear Reduction System Test Rig

To overcome the problem, a solid feed system was developed which consisted of a fully fluidized bed of polystyrene spheres of approximately 0.008 in. to 0.0016 in. diameter into which the alumina particles were fed. The alumina particles formed aggregates of larger particles which were heavier than the plastic beads and therefore would sink to the bottom of the bed. The vigorous motion of the fluidized plastic beads would impact and rub on the alumina aggregates to free the particles from the aggregate surface. Once the individual alumina particles were separated from the aggregate they became fluidized and were lifted out of the bed by the air stream. Although the above described process was not actually observed, it appeared to be a logical explanation of the slow release rate of the solid particles from the bed after a batch of alumina particles was added to the bed. The solids discharge rate from the bed was initially high and gradually decreased with time. Bench tests showed that over a 40-minute period from the time a batch of solids was added, the solid discharge rate decreased gradually from 100 percent to 25 percent of the initial rate in a consistent and repeatable manner. For the purpose of the comparative wear rate tests in the test rig, the repeatability of solids feeding to the test cylinder over the test period is more important between one test and another than achieving constant feed rate. Therefore, this fluidized feed system was accepted for the test rig.

7.1.3 Exhaust System

Excess air and alumina particles supplied by the feed system were allowed to discharge from the test cylinder through a 1.75-in. pipe to a 5900-cubic-inch receiver, where some of the particles and liquid were separated out. The remaining air/alumina/water mixture was vented through a standard chemical hood.

The size of the line and the volume of the receiver were arrived at by trial and error to maintain pressure fluctuations in the test cylinder due to piston motion at a level that would not upset the consistency of delivery of the particle feed system. This pressure fluctuation was ± 10 inches of water column at 1,000 rpm. The average pressure was maintained slightly above 10 inches of water so that the minimum system pressure stayed above atmospheric pressure. This was done so that the collected liquid could be drained during the test.

7.1.4 Test Cylinder Lube Oil System

While the lower part of the test rig was lubricated by the lube oil system, which is a part of the engine, the test cylinder had to be lubricated as well. This was accomplished by a simple gravity system, as shown in Figure 7.1. Oil was introduced at several points on the circumference of the liner at points near the bottom of the piston skirt with the piston at top dead center. The top piston ring and the liquid piston ring did not override the lube oil ports.

Details of the test piston, labyrinth, and piston ring arrangement are given in the following section.

7.2 TEST CONDITIONS

Four comparative tests were selected for the single-cylinder test rig, the results of which provided the necessary information to draw some preliminary conclusions on the technical feasibility of the liquid piston ring concept. These four tests were as follows:

- Conventional piston ring without liquid ring above, air with alumina particles - baseline case.
- Liquid piston ring with water, air with alumina particles.
- Liquid piston ring with water and surfactant, air with alumina particles.
- Liquid piston ring with water and surfactant, no alumina particles in air.

Specific test conditions selected for the test rig are described below.

7.2.1 Liquid Piston Ring

Based on the observation of flow patterns from the bench simulator tests, the 90° saw-tooth labyrinth was selected for the liquid piston ring test in the single-cylinder rig. The tip clearance of the labyrinth was chosen to be 0.006 in. so that at 1,000 rpm and 4.33-in. stroke the Reynolds No. would be comparable to the medium-speed engine with a 0.0025-in. labyrinth tip clearance and 10.5-in. stroke operating at 1,000 rpm. The dimension of the labyrinth is shown in Figure 7.3. The height of the labyrinth section was 1.50 in. with 7 teeth. The Reynolds number, Re_a , in the test rig was about three times that in the simulator, which was $Re_a = 6,600$. This should have provided higher turbulence in the labyrinth cavities and more effective scrubbing of the wall.

7.2.2 Solids Particle Feed Rate

Based on the ash loading of 0.00251 lb/bhp-hr for the medium-speed engine described in Section 5.3, the total ash loading per cylinder at 1,000 rpm and 257 hp was 0.645 lb/hr. Since wear was a function of ash density on the cylinder wall, the relevant parameter to simulate was the average ash loading per unit cylinder surface area. Assuming 10 percent of the ash entering the cylinder deposited on the cylinder wall, the ash deposition rate per unit cylinder wall area was 0.000217 lb/hr-in². For the same deposition rate in the test rig of 3.44-in. bore x 4.33-in. stroke, the solid feed rate to the cylinder should be 0.102 lb/hr (46 gm/hr), again assuming 10 percent of what entered the cylinder deposits on the wall. The solid feed rate was established prior to the test by setting the air supply pressure at 50 psig and adjusting the micrometer valve at the inlet to the fluidized bed plenum. However, it was found after the test that the solid feed rate was

somewhat higher than the set rate. The explanation for this deviation was that the pressure fluctuation in the fluidized bed resulting from cylinder pressure fluctuations caused cyclic variation of flow rate in the bed and consequently a higher rate of solids discharge.

7.2.3 Liquid Piston Ring Flow Rate

Since the engine fuel pump was used to inject the liquid near the top dead center position, the liquid feed rate to the ring was limited by the maximum capacity of the fuel pump. At 1,000 rpm, the mean injection rate was approximately 2,700 ml/hr (or 5.94 lb/hr). At a solid deposition rate of 0.0102 lb/hr, the average concentration of solids in the liquid ring was 0.17 percent. This was considered a reasonably low concentration to avoid settling of the particles due to a change in the flow characteristics of the slurry.

7.2.4 Water/Surfactant Mixture

The surfactant used for the test was DeSonic LFB-65 supplied by Deseto, Inc. The amount of surfactant added to the water was established through bench testing. The test procedure was to wipe the inside surface of a glass test tube with a thin layer of oil, dust a small amount of ash particles onto the surface, partially fill the tube with a prepared solution of water and surfactant, and shake the tube and observe the ash captured on the oil film. The test showed that with 1 percent or higher concentration of the surfactant, the ash and oil film were effectively removed from the glass surface. Therefore, a water solution of 1-percent surfactant was selected for the last two tests.

7.2.5 Top Piston Ring Design

The differences between the conditions in the test rig and in a conventional engine are the temperature and pressure in the cylinder and the conditions the top piston ring is subjected to during an engine cycle. In the conventional engine, the piston ring is pressure-energized against the bore to effect the sealing of the high pressure gas in the cylinder. The end gap of the piston ring causes only a small amount of gas leakage. For the test rig, the cylinder pressure was only slightly above atmospheric. Therefore, an "O" ring was used behind the piston ring (see Figure 7.3) to provide the necessary force to keep the ring in contact with the bore. However, the contact pressure of the ring on the bore was significantly lower than that in a conventional engine during the combustion process. The "O" ring also served the function of sealing the liquid from leaking through the path behind the piston ring. In the case with a liquid piston ring, the end gap of the piston ring would cause the liquid to leak into the crankcase. To minimize this problem, two

piston rings, one on top of the other, were used with the gaps of the two rings offset 180 degrees. It should be understood that although this piston ring configuration provided a comparative test for the different conditions, it by no means represented the optimum design for each situation. More detailed discussion will be given later on this subject.

7.2.6 Cylinder Pressure

The average cylinder pressure was set for all tests at 10-in. H₂O by adjusting the vent valve downstream of the exhaust receiver. The pressure was set above atmospheric pressure to assure drainage of water from the slurry collector. As mentioned earlier, the test rig was not designed to duplicate the operating temperature and pressure conditions in a real engine. The receiver and exhaust pipe were sized to reduce the compression ratio of the test rig to minimize the pressure fluctuation in the cylinder due to the compression action of the reciprocating piston. The measured pressure swing in the cylinder was reduced to ± 10 -in. H₂O from the average.

7.3 TEST RESULTS AND DISCUSSION

There were a total of five test runs, of which the first two were under the same baseline condition without the liquid piston ring. In Test 1, the labyrinth section for the liquid ring was attached to the top of the piston. After the test, it was felt that the labyrinth section may have caused a paste mixture of alumina and oil to collect in the labyrinth section and contribute to the wear of the piston ring and liner. In Test 2 the labyrinth section was removed and the piston was more representative of a conventional design. Test 3 was the water ring test with alumina feed. Test 4 was the same as Test 3 except surfactant was added to the water. Test 5 had the same liquid as in Test 4 with the alumina feed.

The duration of the tests varied between 0.5 to 1.67 hours. The duration was limited to this short time, because the wear rate of the piston ring and liner was so high that additional testing time would cause the piston ring to lose the contact pressure exerted by the "O" ring.

During the test, measured quantities of alumina particles were added to the fluidized bed at fixed time intervals. The total bed material was weighed before and after the tests to determine the total amount of solids fed to the test cylinder.

Liquids were collected and measured in three locations; namely, the liquid slurry collector, the exhaust receiver, and the oil collector. (See Figure 7.1.) The unaccounted liquid was assumed to have discharged with the air.

All the piston rings and the cylinder bore were measured before and after the test. The wear of the piston ring was expressed in terms of the increase in end gap when the O.D. of the ring is held constant. Liner wear rate was expressed in terms of radial increase, averaged over the circumference of the bore and four points along the axis of the bore.

The results of the wear tests are summarized in Table 7.1.

7.3.1 General Inspection

After completion of each test, the cylinder was disassembled for inspection. The following are the specific observations made for the different cases.

- In the baseline case, a relatively thick mixture of oil and solid particles accumulated on the entire outer surface of the piston. See the photographs in Figures 7.5 through 7.8. In Test 1, this mixture also accumulated in the labyrinth section above the piston.
- The thick mixture of oil and solids became diluted at the lower part of the cylinder where fresh oil was injected. The diluted mixture flowed down and drained into the oil collector. In an actual engine, this mixture would be drained into the crankcase.
- With the water ring, the liner and piston surfaces were relatively clean, but a mixture of oil and solids was still present here and there. Because water was mixed into the oil, the solid/oil/water mixture appeared to be more fluid. Photographs of the components are shown in Figures 7.9 through 7.11.
- Water collected in the slurry collector contained solids and oil. This was an indication that solids and oil were scrubbed by the water ring.
- When surfactant was added to the water, the piston and cylinder surfaces were practically free of solids except in the upper part of the cylinder above the travel of the labyrinth section. The liquid collected from the various points contained suspended solids. In this test, a significant amount of liquid leaked past the piston rings and was collected in the oil drain. Photographs of the components after the test are shown in Figures 7.12 through 7.14.
- The solids generally appeared black instead of the white for the feed alumina. This was due to wear material from the piston ring and the cast-iron liner.

TABLE 7.1
WEAR DATA FOR THE LIQUID PISTON RING TEST RIG

Liquid Piston Ring Test Rig Data:					
Test Number	1	2	3	4	5
Bore, (inches)	3.4375	3.4375	3.4375	3.4375	3.4375
Stroke, (inches)	4.33	4.33	4.33	4.33	4.33
Liquid Piston Ring	No Ring	No Ring	Water Ring	1% Surfactant 99% water	1% Surfactant 99% water
Liquid Feed Rate, (c.c./hour)	N/A	N/A	2556.67	2707.74	2790.00
Speed of Engine, (RPM)	1000	1000	1000	1000	1000.00
Run Time, (hours)	1.00	0.50	1.50	1.67	1.50
Rate of Oil Feed into Cylinder, (c.c./hour)	1.62	1.80	1.74	1.80	1.80
Alumina Particle Feed Rate, (grams/hour)	60.15	54.96	89.71	73.71	0.00
Percent of Liquid in Collector, (%)	N/A	N/A	45.24	27.48	52.83
Percent of Liquid through Exhaust, (%)	N/A	N/A	45.84	28.20	37.47
Percent of Liquid Leak by Piston Ring, (%)	N/A	N/A	8.95	44.32	9.70
Liner Wear Rate, (inches/hour)	0.003685	0.003945	0.002935	0.004382	0.000060
Piston Rings Wear Rate, (inches/hour): (as measured by end gap increase)					
Top First Ring	0.186	0.142	0.081	0.074	0.003
Top Second Ring	0.178	0.136	0.090	0.069	0.003
Second Ring	0.164	0.136	0.077	0.072	0.002
Third Ring	0.193	0.158	0.103	0.098	0.003
Oil Wiper Ring	0.196	0.248	0.127	0.148	0.009

Figure 7.5 Test Components After Test 1

Figure 7.6 Test Components After Test 2

Figure 7.7 Test Components After Test 2

Figure 7.8 Test Components After Test 2

Figure 7.9 Test Components After Test 3

Figure 7.10 Test Components After Test 3

Figure 7.11 Test Components After Test 3

Figure 7.12 Test Components After Test 4

Figure 7.13 Test Components After Test 4

Figure 7.14 Test Components After Test 4

- When solids were not introduced and surfactant was added to the water, the piston and liner were clean. The liner surface appeared polished as opposed to the dull appearance after all other tests.
- In all tests with solids, the piston rings were visibly worn. The amount of wear on the radial thickness of the piston rings was uneven around the circumference. This could be a result of uneven deposition of particles due to the port arrangement or the unevenness of the radial pressure applied from the "O" ring behind the piston rings.

7.3.2 Piston Ring and Liner Wear

The wear rates of the piston rings and cylinder liner were very high in all cases when solid particles were introduced to the cylinder. Tests 1 and 2 were different in that the labyrinth section was attached to the piston top in Test 1 and not in Test 2. These results indicate that the presence of the labyrinth section and the accumulation of oil/solid mixture in the labyrinth section did not contribute significantly to the wear of the parts.

With the liquid ring, the wear rate was reduced by a factor of 2, as shown by comparing Tests 3 and 4 with Tests 1 and 2. This clearly indicates that the liquid ring did remove particles from the wall to a degree sufficient to affect wear. However, visual observation of the parts in the two tests with water and the water/surfactant ring suggested that the mechanisms of how the solid particles became trapped between the piston rings and the liner wall were different. In Test 3, the water ring helped to reduce but not eliminate the oil/solid mixture on the surfaces. The presence of the solids on the wall would suggest that the mechanisms of wear in the case of water ring were similar to the baseline case. As the piston ring slides on the liner surface, it rides over the oil/solid layer as it is supposed to do in a normal engine, thus causing the solids to be pushed and trapped between the rubbing surfaces. The reduced wear rate with the water ring reflected the reduced amount of solids on the wall.

For the case with the water/surfactant ring (Test 4), the surfaces were free of alumina particles, and therefore the piston ring would not slide over the particles suspended in water. A reasonable description of the mechanism was that the leakage of the liquid past the piston ring caused the suspended alumina particles to be accumulated in front of the ring and eventually run over by the ring. Trapping the particles between the piston ring and the liner surface would open up the gap and cause more liquid to leak, as evidenced by the large fraction of liquid collected from the oil drain, and would consequently bring more particles to the front of the piston ring.

The photographs of the piston ring surface for the various cases are shown in Figure 7.15 through 7.17. They all show similar wear marks parallel to the piston axis.

Test 5 was designed to simulate the ideal condition in which a water ring with surfactant addition was established above the piston ring such that the solids contaminated portion of the liquid ring was kept near the top of the liquid ring and the piston ring contacted only the clean water/surfactant mixture. The test showed a potential reduction in wear by a factor of 50 over the baseline case. The water leakage past the piston rings was significantly different in Test 5 (compared with Test 4). This supported the postulated mechanism that the accumulation of solid particles between the piston ring and the cylinder liner due to leakage of the contaminated liquid caused the leakage to increase further.

Based on the analysis of the test results, we concluded that the liquid piston ring was effective in scrubbing and reducing the amount of particles held by the oil film on the wall. When a proper amount of surfactant was added to the water the movement of the liquid ring during the up-stroke effectively removed from the wall the solid particles that were deposited on the wall during the down-stroke of the piston. The wear of the piston ring and liner under this condition was not the result of the reduced oil film on the wall due to the surfactant. The wear resulted from the leakage of the contaminated liquid past the piston ring, which caused the solid particles to accumulate and become trapped between the piston ring and the liner surface. It became clear that, in addition to cleaning the solid particles off the wall surface, preventing the contaminated liquid from leaking past the piston ring was equally important for the liquid piston ring concept. The liquid piston ring concept for wear reduction in a coal-slurry-fired diesel engine is therefore technically feasible provided that an improved piston seal ring system is developed to prevent leakage of the contaminated liquid past the piston rings.

Since this study has shown the feasibility of cleaning the wall with a liquid ring, the next R&D step should be focused on the improvement of the sealing system to prevent leakage of the liquid past the piston ring. There are two steps in the development of an improved liquid ring system. The first step is to improve the liquid sealing of the piston ring system. The second step is to design a liquid supply system in which the liquid is supplied through the gap between the piston ring and the bore from a buffer zone below the ring. The flow of liquid through the gap will continuously flush the particles away from the zone immediately above the piston ring. With the presence of the liquid ring, the piston ring will not be subjected to high temperature. Special materials including state-of-the-art plastic materials currently used for seals in compressors and pumps can be considered. In recent years, considerable interest has been raised in the subject of high

Figure 7.15 Magnified View of Top Ring Surface

Figure 7.16 Magnified View of Top Ring Surface

Figure 7.17 Magnified View of Top Ring Surface

pressure slurry pumping for pipeline transport of minerals and coal slurries. Advancements have been made in the development of plunger pumps and their seal systems. New technology developed for sealing reciprocating plunger pumps should be applied and modified as needed for the engine system.

8. CONCLUSIONS AND RECOMMENDATIONS

The following conclusions summarize the significant results obtained as the program progressed from modeling, to the flow visualization bench test, to the single-cylinder test rig where actual wear measurements were made.

- Baseline testing in the single-cylinder test rig with alumina particle feed and no liquid ring showed that the particles mixed with the oil to form a solid/oil mixture on the cylinder wall which caused high wear of the piston ring and liner. The solid/oil mixture also leaked past the piston ring into the space below the piston.
- A water ring with a saw-tooth shaped labyrinth effectively reduced the amount of solid/oil mixture on the cylinder wall, thus reducing the wear rate by a factor of 2.
- A water ring with surfactant addition was most effective in keeping the solid particles from adhering to the cylinder wall. However, the piston ring and liner wear rate was comparable to that with the water ring. The leakage of the contaminated liquid with suspended solids past the piston ring caused the solids to be accumulated and trapped between the piston ring and the cylinder wall, thus producing the wear effect.
- Preventing leakage of the contaminated liquid past the piston ring was identified as one of the major factors in the liquid piston ring concept and should be the subject for the next R&D program.
- Based on the flow visualization study on a bench simulator, a liquid ring with a 90° saw-tooth labyrinth provided high shear rate at the cylinder wall and constant agitation in the labyrinth channels, which prevented solids accumulation.
- The once-through version of the liquid ring concept is the most desirable from the standpoint of system simplicity. The once-through system was tested in the single-cylinder test rig and the problems identified are not easier to solve with the other versions of the concept.
- Liquid supply rate to the once-through system to maintain low particle concentration is sufficiently low so that no detrimental effects on engine thermal performance would be expected.

- Heat transfer to the liquid ring through the piston and liner must be limited to prevent boiling away the liquid ring. Conceptual piston designs for minimizing this heat transfer are presented.
- A dynamic analysis of the liquid ring including viscous drag, inertia, and labyrinth pressure drop effects was carried out which showed large static pressure variations in the liquid ring during a complete engine cycle. These pressure variations could allow significant loss of liquid in the ring if heat transfer was not limited.
- The pressure fluctuations in the ring were shown to be a strong function of ring height and engine speed, and a weak function of the liquid flow rate to the labyrinth and the number of labyrinth teeth when the labyrinth clearance is small.

Based on these results and conclusions, the following steps could be taken to continue exploration of the liquid ring concept.

- Select several concepts for reducing liquid ring leakage past the top ring. Establishing a liquid buffer zone with controlled leakage to the labyrinth seems most promising.
- Test one or more concepts in the single-cylinder test rig.
- Design a bench-scale piston incorporating the liquid piston ring and reduced heat transfer concepts.
- Conduct firing tests of the liquid ring concept in a single-cylinder test rig.

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