Investigation into the relationship between knock intensity and piston seizure

by J. C. Fitton* and R. J. Nates**

ABSTRACT

This work concerns an investigation into the role of knock on piston seizure in spark-ignition engines. The scope of the research includes a theoretical evaluation of the relationship between knock intensity and ring gap closure, together with the subsequent fatigue and failure of the piston land. An experimental technique is developed to determine the ring gap as a function of knock intensity under operating conditions in a CFR research engine. The results of both the theoretical and experimental study show that knock intensity has a significant effect on ring gap closure. From this, it is evident that knock produces disturbances in the quench zone, resulting in a general increase in the heat flux to the combustion chamber surfaces. The resulting thermal expansion of the ring is sufficient to cause fatigue failure of the piston second land. An exponential reduction in the piston life is predicted when moving from light to heavy knock conditions. This is particularly problematic under high speed knock conditions where the operator would be unaware of any problem until catastrophic failure. This work was performed as a final year BSc (Mech Eng) project at the University of Cape Town.

Introduction

Knock and associated knock damage has been observed in spark-ignition engines since the early days of engine development, as early as 1882[1]. Since knock limits engine performance and can lead to serious damage, numerous papers have been published contributing to the explanation of this phenomenon.[2] Most of this work has centered around methods to prevent the onset of knock, without much attention being paid to the mechanisms of knock damage. The rationale being that any knock whatsoever is detrimental to engine durability.

It has however been found that light knock has no measurable effect on engine durability[3]. With this in mind, it would be of great value to gain a quantitative understanding of the margin of safety for the transition from non-knocking to knock damage conditions.

Knock and knock damage

Knock is caused by the auto-ignition of the compressed end-gas in front of the deflagration flame-front. This auto-ignition can result in a series of shock waves which propagate from the auto-ignition site and are reflected around the combustion chamber[4]. These waves set the combustion cavity in resonance which in turn causes the engine structure to resonate with the audible pinging or knocking sound.

The quantitative measurement of knock intensity is controversial, in as much as different researchers have adopted various methods for monitoring this parameter. A review of all these procedures is beyond the scope of this work and therefore the most commonly used knock intensity parameter was adopted for this work. In this paper, knock intensity is defined as the maximum measured amplitude of the pressure fluctuations inside the combustion chamber[5]. The knock intensity is measured by means of digitizing the pressure trace and filtering out the low frequency spectrum such that the pressure fluctuations of the knock signal may be measured. The maximum amplitude of this signal is then taken to indicate the knock intensity.

The mechanism of knock damage is not fully understood, but manifests itself in two main forms, as follows:

Surface Erosion:

Areas of the aluminium piston crown and top-lands (and in some cases the head and gasket) are eroded in the end-gas region. The damage seems to be a result of the localized nature of the initial auto-ignition, forming destructive detonation waves in the end-gas zone[6, 7]. This type of damage is rarely observed with cast iron pistons.

Piston Seizure:

General piston overheating and seizure occurs as a result of the resonating shock waves disturbing the insulating quench zone at the surfaces of the combustion chamber, resulting in increased heat flux into the piston and head[8, 9]. This can lead to the compression rings or piston lands expanding and seizing in the bore.

Surface erosion, while damaging, is not catastrophic to engine operation, although under severe knocking it can result in pre-ignition and a runaway situation could develop. The pre-ignition effectively advances the ignition timing causing an increase in knock intensity. Erosion type damage is currently being used as a means of developing a damage threshold for fuel and engine characteristics[10].

Little work has been performed in the case of piston seizure as a result of knock. The safety margin between non-knocking and piston seizure is of particular importance when considering the condition of high-speed knock. At high speeds, the frequency of the general engine noise is such that it masks the sound of knock, thus the operator is unaware of any potential damage until catastrophic failure occurs.

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The aim of this work is to investigate the relationship between knock intensity and piston seizure.

The term "piston seizure" in fact includes many different modes of failure. The most common form of catastrophic engine failure is found to be the breaking of the piston lands due to the development of fatigue cracks at the inside corner of the ring grooves as shown in figure 1. Betts[11] has shown that this is a typical failure mode for engines running at high speed under knocking conditions. This failure is as a result of the overheating of the rings, causing the ring gap to close, in turn setting up hoop stresses and the eventual binding of the ring in the bore. The ring is then dragged against the bore producing a cyclic bending moment on the piston lands. Under these conditions the piston rings will eventually cause complete seizure and in many cases the piston lands will be found to be broken, aggravating the seizure damage.

The approach of this work was to determine a quantitative relationship between knock intensity and ring gap closure.

A measure of the effect of knock on this failure mode is therefore the amount of ring gap closure as a function of knock intensity.

Theoretical development

In order to interpret any measured data it was deemed important to develop a theoretical understanding of the effect of knock intensity on ring seizure. Numerical models are being developed for the predition of heat flux as a function of knock intensity[12], but as yet these are only applicable to specific engine geometries and operating conditions, and in most cases do not include the condition of the piston rings.

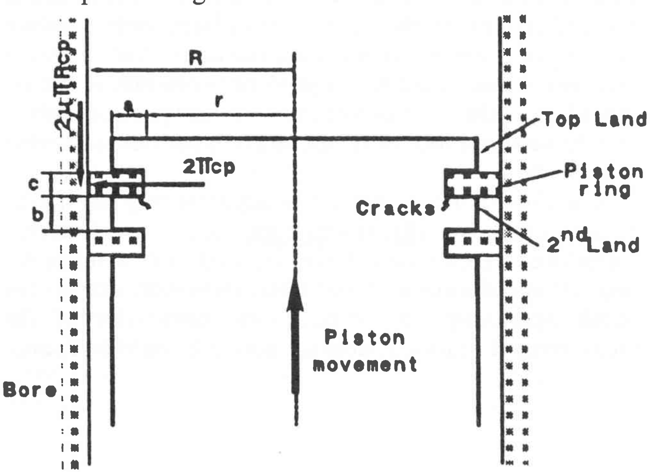

Figure 1 – Vertical section through piston showing crack sites.

As this work is a first attempt at developing a relationship between knock intensity and piston seizure, it was decided to use existing measured data to describe a bulk ring temperature as a function of knock intensity.

French and Atkins[13] recorded piston temperatures (at various locations) as a function of spark advance for knocking and non-knocking conditions. These tests were performed under steady state conditions, wide open throttle, with two different octane fuels. Typical results for the temperature at the surface of the top piston land are shown in figure 2.

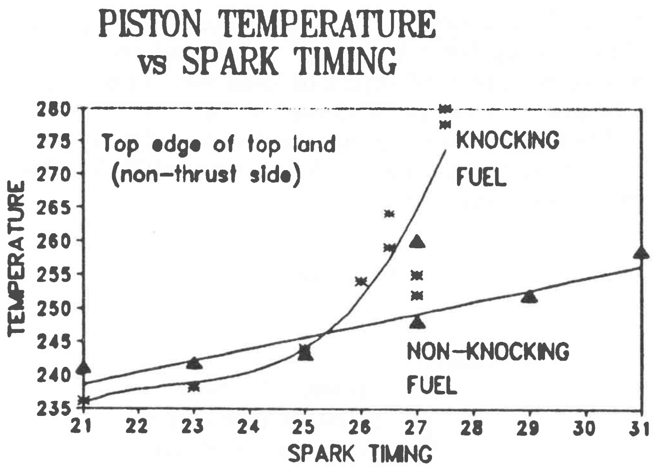

Figure 2 – French & Atkins data depicting the effect of knock on piston temperature.

As a first estimate, it was assumed that the rate of change of the ring groove temperature with respect to spark advance is equivalent to the change experienced in the bulk ring temperature. From a knowledge of the thermal expansion coefficient for the ring material, the magnitude of the ring expansion could then be determined as a function of spark advance from:

$$\frac{\Delta l}{\Delta SA} = l\alpha \frac{\Delta T}{\Delta SA} \qquad (1)$$

Where

l = *Ring length (m)*
SA = *Spark advance (°CA)*
α = *Coefficient of thermal expansion*
T = *Ring temperature (°C)*

The ring gap calculated from these data (using a typical value for cold ring gap) is shown in figure 3.

Seizure would not occur at the point of ring gap closure, but further expansion would result in increased circumferential and radial stresses being produced in the ring. In turn, this would result in increased pressure between the ring and bore, breaking down the lubricating

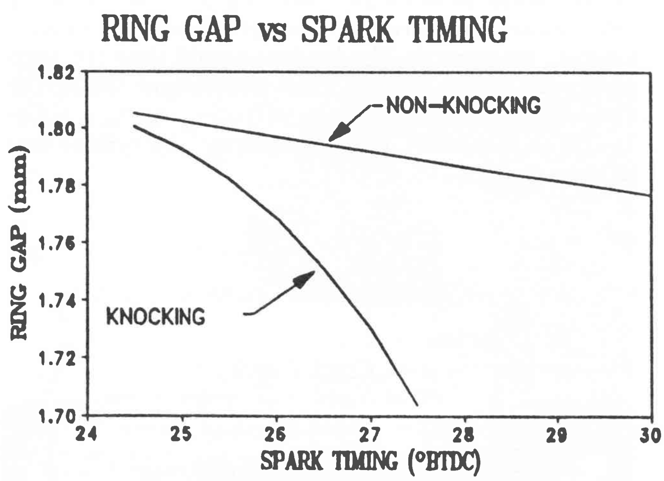

Figure 3 – French & Atkins manipulated data to determine the effect of spark timing on ring gap closure.

film and increasing the frictional force at the interface. This binding force would be transferred to the ring groove in the form of a cyclic bending moment producing a fatigue situation in the corners of the groove.

The radial pressure (p) may be determined by approximating the system to a shrink fit situation[14] described by the following relationship:

$$a\ ET = -p\ \frac{(R^2 + r^2)}{(R^2 - r^2)} \qquad (2)$$

where

E = *Young's Modulus (190 GPa for cast iron)*
R = *Outer radius of ring (m)*
r = *Inner radius of ring (m)*
p = *Pressure at interface (Pa)*

The shear force acting at the ring-bore interface may be estimated by assuming a value for the coefficient of friction. A typical value of 0,1 for near metal to metal contact was chosen[15]

Using a two dimensional bending moment model, as illustrated in figure 1, the stress (σ) induced at the inner groove radius may be determined from:

$$\sigma = \frac{(2\pi Rc)\ \mu ab}{2I} \qquad (3)$$

where

μ = *Coefficient of friction*
a = *Depth of ring groove (m)*
b = *Height of piston land (m)*
c = *Height of ring (m)*
I = *2nd Moment of area of piston land (m^4)*

It is assumed that the second compression ring has not seized and caused negligible bending on the underside of the second piston land. This is a less severe condition which would yield a more optimistic estimate of the piston life.

During the upward motion of the piston, the inner groove radius would experience negligible stress, while during downward movement the stress would be as determined in equation 3. The loading would thus cycle between zero and this stress level. The fatigue life of the piston land could now be estimated using the Paris equation[16] (equation 4), piston geometry and typical material properties.

$$\frac{da}{dN} = \int_{a_i}^{a_{cr}} A(\sigma\sqrt{\pi a})^m \qquad (4)$$

where

a = *Crack length (mm)*
N = *Number of cycles*
a_{cr} = *Crack length at failure*
a_i = *Initial crack length (1 μm)*
A = *Crack growth rate (mm/cycle)*
m = *Paris Law Exponent*

Typical values for "A" and "m" for piston alloy are $45{,}6.10^{-10}$ and 3,0 respectively.

With a knowledge of the relationship between ring gap or ring temperature and knock intensity, this analysis could be used to estimate the time to failure under particular knocking conditions. The aim of the experimental work was to provide this relationship.

Experimental procedure

A CFR Waukesha engine was used at a constant speed of 890 rpm and compression ratio of 7,5:1, under steady state conditions. Oil and coolant temperatures were monitored and held constant.

Knocking combustion was induced and its intensity varied by means of altering the ignition timing. Both 93 and 102 octane (RON) gasoline was tested, so that the effect of spark-advance which itself causes an increase in surface temperatures, could be eliminated. The 102 octane fuel did not knock under any of the operating conditions.

The pressure trace was monitored using an AVL 12QP peizoelectric pressure transducer and a Gould 420 digital storage oscilloscope. The knock intensity was determined using the maximum pressure amplitude recorded through a high pass filter. This value was monitored during each test run.

As discussed in section 3, it was important to measure the effect of changes in knock intensity on changes in the bulk ring temperature. A J-type thermocouple was embedded behind the top compression ring for this purpose. In order to establish that steady state conditions had been achieved a continuous temperature reading was required. Thus, the thermocouple wire was led onto the connecting rod and then out of the crankcase in a large loop to reduce bending of the wire. This system rarely allowed for more than two steady state readings to be taken before breaking. Due to the time constraints on this project, only a very limited number of temperature measurements were thus made.

A method of measuring the steady state ring gap had to be developed. Initially, it was hoped to use an electronic transducer which would have required only a single fitting for the duration of the tests. However, due to the harsh operating environment and sensitivity of the measurement required, nothing suitable could be found.

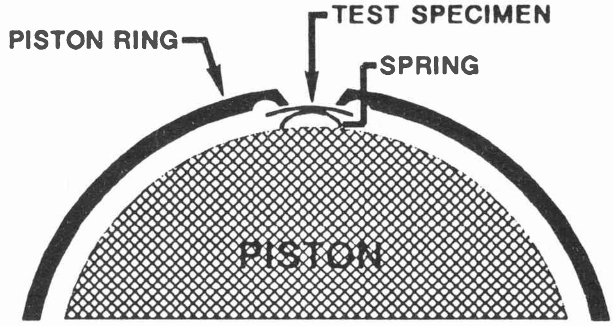

Figure 4 – Horizontal section through piston depicting the scribes, aluminium specimen and spring.

Instead a mechanical system was developed which necessitated the removal of the piston between each measurement. The inside surfaces of the ends of the top compression rings were machined into sharp scribes. An aluminium plate (0,9 mm thick) was placed behind the scribes as shown in figure 4. A curved spring steel plate was placed behind the aluminium plate inside the ring groove so as to force the plate against the scribes.

As the ring gap closed during engine operation, two witness marks were made on the aluminium by the scribes. After completion of a test run, the plate was removed and the minimum ring gap was measured using a graduated microscope. A typical plate with scribe marks is shown in figure 5.

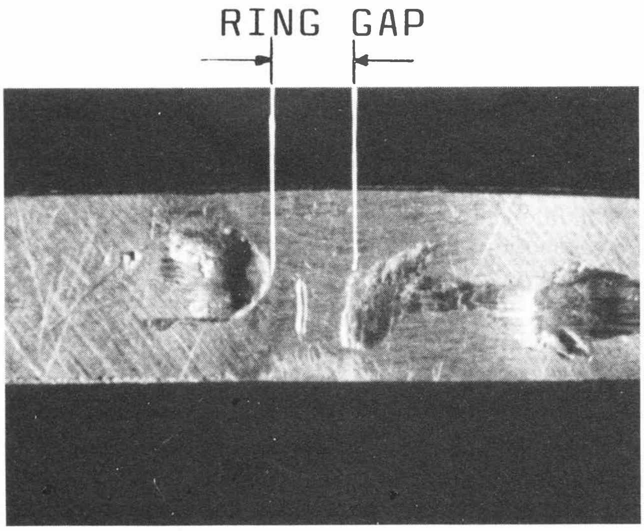

Figure 5 – Ring gap test specimen with witness marks.

The installation of the modified ring, plate and spring had to be performed with great care, so as not to cause spurious witness marks to occur during the fitting.

Results and discussion

Figure 6 shows the experimental relationship between ring gap and ignition timing for both knocking and non-knocking fuels, as determined on the CFR engine. From the results of the non-knocking gasoline, the effect of spark-advance is shown to reduce the ring gap. As timing is advanced, combustion occurs at higher pressures, thus the combustion chamber temperatures are increased, increasing the bulk ring temperature.

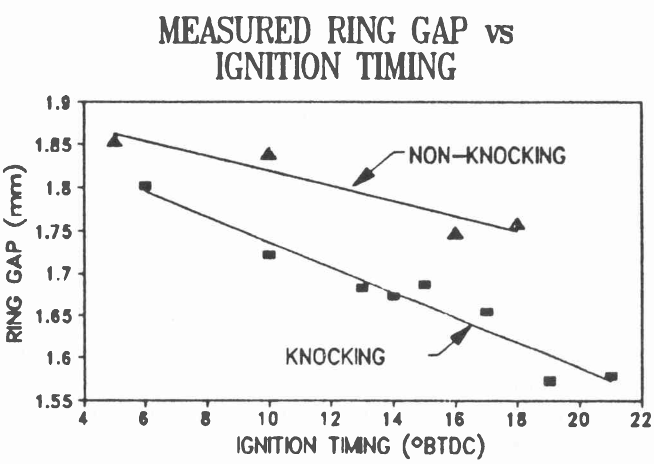

Figure 6 – Effect of knock intensity on ring gap closure.

The relationship between piston surface temperatures and ignition timing for normal combustion has been shown by various authors[3, 13] to be approximately linear. Therefore, only a few tests were performed on the high octane fuel, and a linear regression was applied to the data (as shown in figure 5).

It is seen that the rate of change of the ring gap with respect to spark advance is much greater for the tests performed under knocking conditions. With an advance in ignition timing under already knocking conditions, the end-gas is subjected to higher temperatures and pressures, and thus the resulting knock is of a higher intensity. This causes a greater disruption of the insulating quench zone and more turbulence at the metal surfaces, resulting in greater heat flux into the combustion chamber surfaces.

The quantitative difference between these two curves is the net effect of increased knock intensity. Figure 7 shows the effect of knock intensity alone on ring gap (the effect of ignition timing having been subtracted). The magnitude of the ring gap closure of 0,1 mm from non-knocking to heavy knock is significant when considering that typical cold ring gaps are only about 0,3 mm (for 80 mm bore). The term cold ring gap refers to the measurement performed when fitting the ring into the bore, therefore the normal operating (hot) ring gap is considerably less than this value.

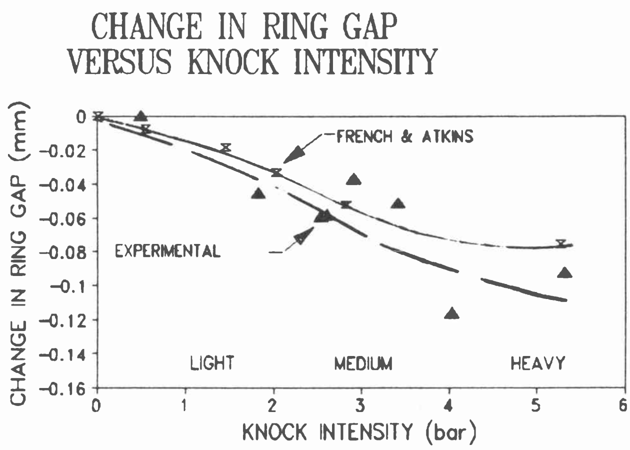

Figure 7 – Effect of knock intensity on ring gap closure.

The calculated change in ring gap using the procedure outlined in section 3 and the data from French and Atkins[13], is also shown in figure 7. Due to the fact that these authors did not measure knock intensity, but merely used the terms no-knock, borderline knock, light-, medium- and heavy knock, equivalent knock intensities had to be assigned to these data. Although this may introduce an appreciable error, the characteristic form of this curve is of interest. It may also be noted that the range of knock intensities tested on the CFR engine conformed to the range "no-knock" to "heavy knock", as was the case in the French and Atkins study.

A second order effect can be observed in the data, with two points of inflection. When moving from non-knocking conditions to borderline knock, a small reduction in

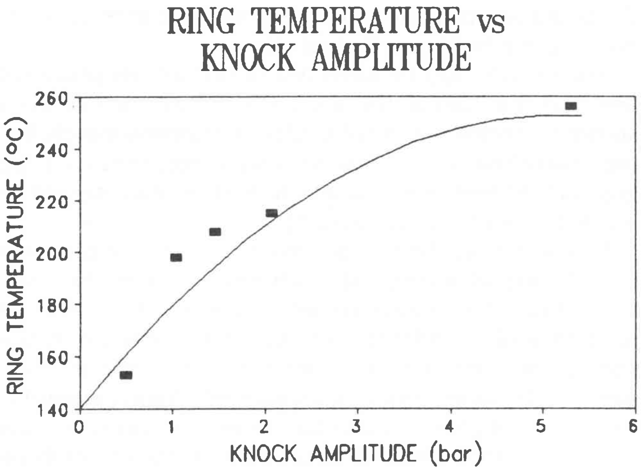

Figure 8 – Measured ring groove temperature versus knock intensity.

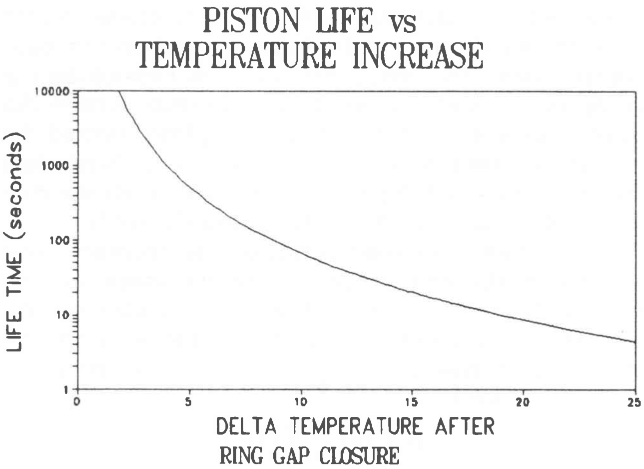

Figure 9 – Piston life to failure after ring gap closure has occured.

ring gap is noted. This is probably a result of the low intensity shock waves disturbing the laminar boundary layer, producing transitional flow (mixed laminar and turbulent) heat transfer.

At the transition from borderline to light knock, an increased rate of change in ring gap is noted. With further increase in knock intensity, the resulting reduction in ring gap is less severe than the initial transition from normal to knocking combustion. This trend may be understood in terms of the effect of knock intensity on surface temperatures.

Figure 8 shows the measured relationship between ring temperature and knock intensity. The transition from borderline to light knock produces a greater temperature increase than at higher knock intensities. If the effect of knock intensity is to alter the local gas velocities (and thus the local Reynolds Number) then from the Reynolds-Colburn Analogy for turbulent flow over flat plates[17], the heat transfer coefficient is directly proportional to the Reynolds Number to the power of less than unity. Furthermore, there is no significant difference between the gas temperatures for normal and knocking conditions. Therefore with the gas temperatures nearly constant and the heat transfer coefficient a power function of Reynolds Number, it is expected that the surface temperatures will follow this power law as a function of knock intensity. Thus the trends observed in figures 7 and 8 could be a result of the effect of knock intensity on local gas velocities.

It is also observed that the magnitude of the ring gap values calculated from the French and Atkins data is less than that for the experimental values. One of the reasons for this is the fact that the latters' tests were performed on an aluminium piston, which would show a higher conductive heat flux than the cast iron piston of the CFR engine used. This would cause the cast iron piston to operate at higher surface temperatures for a particular heat flux, resulting in smaller ring gaps.

In general, the calculated values compare favourably to the limited experimental observations. The fact that the second order effects are not easily observable in the measured ring gap data can be attributed to the following considerations:

1. Only a limited number of tests were performed.
2. The ring gap measured is the minimum for that particular test run. Care must therefore be taken to ensure that the engine is gradually brought up to operating conditions. It is possible that in some of these tests an overshoot of temperature occurred before being controlled to the desired conditions.
3. It is possible that during fitting the scribes caused spurious witness marks to occur. The cold ring gap should be increased and the tension in the ring squeezer should be as low as possible so as to eliminate this possibility.

From the fatigue calculations described in section 3, an estimate of the piston life as a function of increase in bulk ring temperature could be made. It must be remembered that the failure criterion used here is the breakage of the piston land, which usually results in complete engine seizure within a few cycles due to the wedging of the land between the piston and bore. Even if this does not occur, intense blowby of combustion gasses would result in localized heating, resulting in the eventual seizure of the piston itself due to thermal expansion.

The life of failure is evaluated in terms of bulk ring temperature changes after the ring gap has closed. This is due to the dependence of the normal hot ring gap on the initial cold ring gap (which in turn is dependent on the particular engine under consideration). A relationship for a typical engine operating at 4 000 rpm (high speed knock region) is shown in figure 8. For borderline knock conditions, the life of the piston is relatively unaffected. However, even at light knock the piston life becomes finite and is rapidly reduced as the knock intensity is increased (ie: ring temperature increased).

An example of the order of magnitude for the shear stress of the ring against the bore as a function of bulk ring temperature for conditions after the ring gap has closed is as follows: At the point of ring gap closure, a typical shear stress is about 7 kPa. A further increase of 35°C in bulk ring temperature would result in a shear stress of 580 kPa.

Table 1 shows the estimated piston life expectancy for an engine operating at 4 000 rpm, bore 80 mm and with typical cold ring gap, under various knock intensities. This illustrates the magnitude of the problem of high speed knock.

The simplest remedy to this problem is to increase the

Knock Intensity	Ring gap	Life time (minutes)
Normal	0.11mm	Infinity
Light	Closed	90 000
Medium	Closed	1.1
Heavy	Closed	0.4

Table 1 – Estimated life time to failure of piston land.

cold ring gap. However, doing this would cause an increase in blowby and a reduction in engine performance and efficiency. Therefore, a thorough understanding of the relationship between knock intensity and ring gap closure is necessary in order to design with a sufficient safety factor.

Conclusions

The conclusions drawn from this project may be summarized as follows:

– A means of measuring the ring gap under operating conditions has been developed and tested. The results from this test procedure correlate with calculated values using independent temperature test data.
– Knock has a significant effect on the thermal expansion of the top compression ring causing it to close by as much as 0,1 mm under heavy knock conditions.
– The thermal expansion of the ring under knocking conditions is sufficient to cause a fatigue situation to develop on the piston top land. Under medium to heavy knock conditions this could result in the failure of the land and the eventual catastrophic seizure of the engine.
– From the form of the relationship between ring gap and knock intensity, it is evident that the shock waves due to the knock produce disturbances in the insulating quench zone around the combustion chamber, resulting in a general increase in the heat flux to the chamber surfaces.
– From fatigue life considerations, it is evident that the effect of knock on this mode of failure is significant. This is particularly serious in the light of high speed knock, where the operator would be unaware of the knock occurring.
– The experimental investigation was limited to the conditions under which the ring gap was finite. The theoretical analysis extended this work to conditions of complete ring gap closure, assuming that the thermal expansion of the top ring due to knock alone was responsible for the fatigue mechanism. Under actual operating conditions, it is expected that there would be a significant heat flux into the ring as a result of the increased friction forces, increasing the pressure between ring and bore (and the bending moment on the land). The second ring would also be expected to contribute to the load cycle on the piston land, thus aggravating the situation further.

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