



Article

The Influence of a Piston Ring Coating on the Wear and Friction Generated during Linear Oscillation

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Abstract: The piston group is responsible for contributing to ~50% of the frictional losses of an engine, which ultimately leads to the waste of fuel. This coupled with the fact that gasoline is a finite resource linked to CO₂-emissions, there is an increased demand of higher performance vehicles, which coincidentally further loads the piston ring. As of yet, there are plenty of studies that already study the piston ring's contact with the cylinder liner. However, this study focuses on a cost-effective Schwing, Reib, Verschleiss (SRV) instrumentation that allows to pre-screen lubricants, additives, materials and coatings for their friction, wear and load carrying capacity including scuffing resistance. As a result of the pre-screening conducted outside of engine by using the SRV instrument, it brings us to the following conclusion: the PVD CrN-TiN 1^o Group coating on the piston ring produces the least wear, as well as the lowest coefficient of friction. Moreover, it is concluded that a coating that is based from CrN and TiN allows the piston ring to perform better in engine settings. A continued understanding of the piston-cylinder-contact assembly only helps engineers, scientists and any other stakeholder to improve on the piston ring and cylinder liner interaction.

Keywords: internal combustion engine; compression rings; viscosity

1. Introduction

The purpose of piston rings (PR) is to provide a sealing between the gap of the piston and the cylinder wall, whilst distributing and controlling the oil and transferring the heat along with stabilizing the piston. This is a departure from earlier days, where the sole purpose of the PR was just to seal off the combustion chamber. The PR in the internal combustion engine (ICE) is accountable for a significant amount of friction loss and the optimization of this tribosystem is very important to increase the overall fuel economy of the engine [1]. For example, frictional losses of the piston-cylinder-contact (PCC, where the PR is in contact with the cylinder liner) in heavy duty diesel engines of trucks can contribute to the overall 4–15% of fuel consumption. As a result, billions of liters of fuel are lost to just overcoming the friction in these vehicles [2]. Significant studies have been conducted in this area, such as R. Rahmani et al. explaining that the internal combustion engine will continue as the main 'source of vehicular propulsion for the foreseeable future.' The authors present a study that takes an integrated approach to the frictional losses, as well as discuss the emission losses of the piston-cylinder system. The conclusions of this experiment state that the cylinder liner temperature is critical in mitigating power loss, in addition to reducing Hydrocarbon and Nitrogen oxide emissions from the compression ring [3]. It is important to note that the compression ring is located nearest to the piston head, which is the top of the piston [4].

This paper offers a novel way to investigate the wear and friction generated by the PR oscillating against the cylinder liner using the SRV tester. This is otherwise known as SRV, which translated from German means oscillation, friction and wear. The SRV instrument can be used to closely simulate specific operating conditions relevant for the functioning of the PR in the ICE.

The primary investigation of this paper is: How different piston ring coatings influence the wear and the friction generated from the sliding motion of the PCC. To the authors' knowledge, there are studies that do consider PR coatings and the effect of it on the PCC assembly but it is the novelty of how this investigation was conducted that presents a realistic, cost-efficient way of studying the PCC movement. Ultimately, it is the goal of such screening to suggest possible coatings of the PR in the PCC assembly, so that the PR can be possibly further optimized. By choosing this parameter of study, it offers a viable way the PR can be advanced without a need for a large change in the current settings of the PCC assembly in the engine, making it a feasible option to be considered for more studies. This approach is also suited for development of alternative and low viscosity engine oils.

2. Modeling the Friction of the Piston-Cylinder-Contact during Engine Operation

During operation cycle of 720° in the ICE, the PR passes all lubrication modes on the cylinder liner and will generate different levels of friction [5]. Piston coatings have the potential to reduce friction and/or wear of the piston's sliding components [6] but also to prevent scuffing. To measure reduction in wear and friction of the PCC, two variables are considered: Loss of mass of the PR and coefficient of friction (COF) of the PCC as a function of time. The following explanations provide a basis of why both these parameters are valid to observe.

Figure 1 illustrates the surface of contact between the PR and the cylinder liner. Over the distance: $\Delta x = x_2 - x_1$ the PCC can be assumed to experience mixed friction, due to the combination of pure hydrodynamic lubrication points (where sufficient amount of lubricant exists to separate the two surfaces in contact) and pure boundary friction points (where there is an insufficient amount of lubricant between the two surfaces in contact) [7].

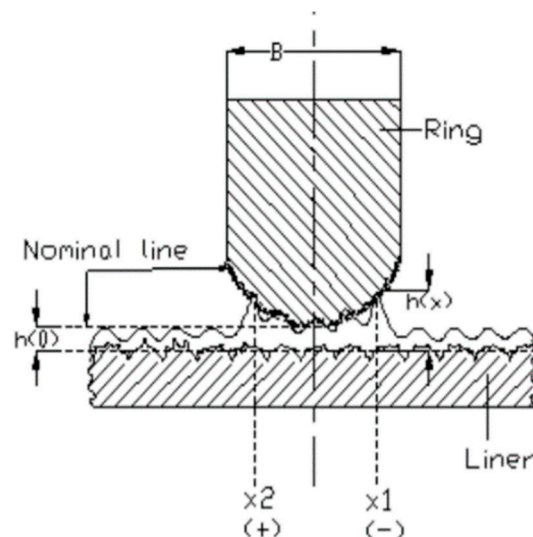


Figure 1. Piston ring (PR) and Cylinder Liner Contact.

Furthermore, scuffing failure can be categorized as catastrophic failure coupled with severe form of wear by substantial material transfer, which can occur during the PCC operation [8]. The location in which the oil film is penetrated is where the sliding on the PR of the cylinder will cause deformation and therefore wear [9].

To extrapolate from these foundational ideas, when there is a penetration of the oil film, the PR will slide over the cylinder liner without sufficient lubrication between the two surfaces. This will cause

deformation and wear. Additionally, the metal to metal contact and the subsequent sliding of it leads to increased friction [10]. This potentially increases the COF leads to higher fuel consumption and lower fuel efficiency. It is particularly difficult to model scuffing failure of normal engine operations due to varying piston sliding speeds during its stroke operations, where the lubrication also varies during engine operation as a function of stroke operation [8]. This speed of sliding motion, temperature and load are simplified in this study. This material transfer that occurs during the sliding of the PCC is studied by the loss of mass of the PR and COF as a function of time. Therefore, a low overall COF, combined with a small loss of mass, is an indicator that it is a well-adjusted tribosystem. The Stribeck-type diagram can be used to model the rubbing of the PR with the cylinder liner in presence of a lubrication, as shown below [11].

From Figure 2, the $\eta V/W$ factor is a lubrication parameter that can be related to the COF through the Stribeck Curve. For small values of $\eta V/W$, in the boundary lubrication region, the COF is high due to the decreased oil viscosity causing metal-to-metal to occur. These two metals then can rub together causing the high COF. In the mixed-film lubrication region, as the load applied decreases or oil viscosity increases, and/or sliding velocity (or linear oscillation) increases, the $\eta V/W$ factor increases. Moreover, the COF decreases until it reaches a minimum, as shown in the mixed-film lubrication region. For higher values of $\eta V/W$, in the fluid-film lubrication region there is negligible asperity contact and COF increases linearly with increasing values of $\eta V/W$ [11].

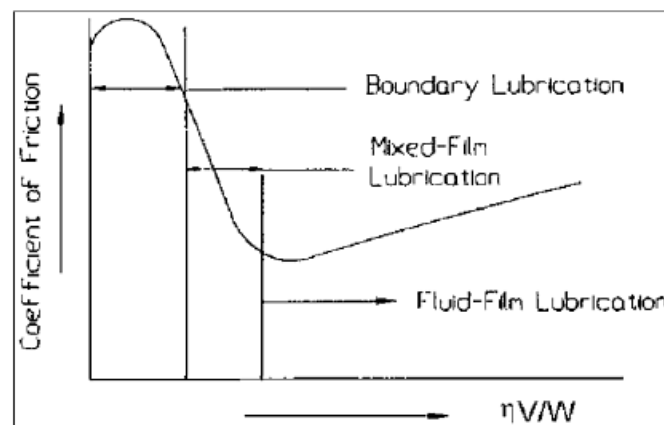


Figure 2. Diagram of Stribeck Curve, η : oil viscosity, V : sliding velocity, W : normal load applied.

The friction generated by the PR is significantly affected by the pressure in the cylinder throughout the four-stroke operation [7]; cylinder temperatures and pressures contribute to corrosive wear [9]. The cylinder is the volume in which the piston moves up and down causing the crankshaft to rotate [12]. These factors are considered during the experiment, so that when the loss of mass and COF of each PR coating are collected, it reflects the PCC as accurately as possible during the operation cycle.

3. Materials and Methods

3.1. Design of Instrument

Original engine parts for the experiment were used, as shown in Figure 3, to replicate the PCC interaction during normal engine operation. All tests are run using a SRV[®] III by Optinol Instruments Prueftechnik GmbH, Westendstraße 125, 80339 München, Germany. The SRV consists of a fixed bottom specimen support and a mobile replaceable top specimen holder. The holder is responsible for clamping the different ring and liner geometries as shown in Figure 4a and is dipped in an oil bath as shown above in Figure 4b.

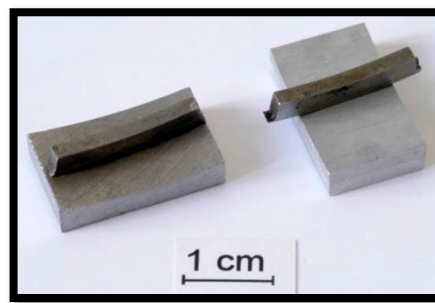


Figure 3. Ready prepared samples of the PR directly from the engine block.

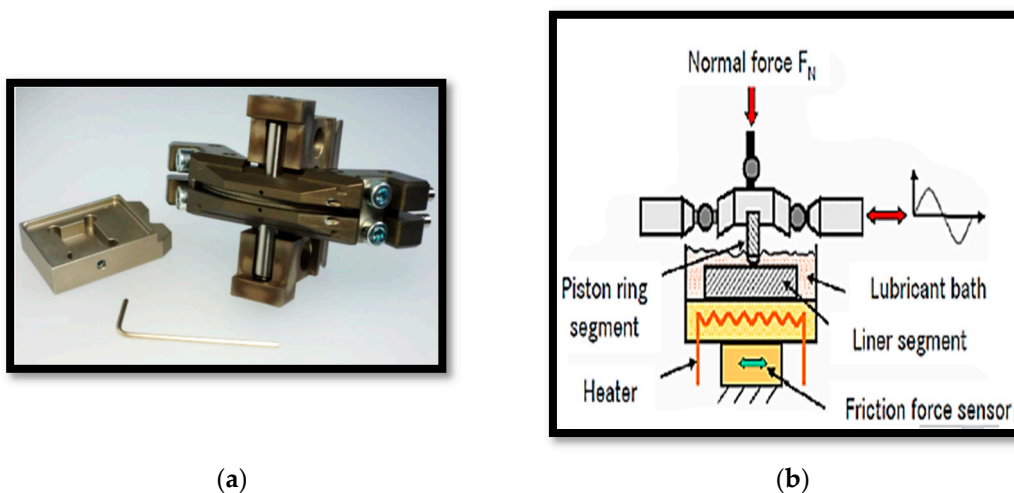


Figure 4. (a) Piston Ring Holder, (b) Schematic of the testing chamber inside the SRV with the upper and lower specimen.

The upper and lower specimens are pressed together by an adjustable normal force and are oscillated tangentially, employing an adequate frequency. This oscillation mimics the motion of the stroke of the PR. The contact area depends strongly on the ring's geometry—such as diameter and cross-section—and the wear of the system. We consider the PCC as the system in this study. The friction force is transferred to the sensor located in the block beneath the test piece, from an applied load range that is varied to mimic the external pressure in a PCC. The head plate is connected to the machine via a single or (two) piezo crystal(s). This/these crystal(s) measure the transverse force created. With the knowledge of both normal and transverse force, the COF can be calculated as their quotient. An oil pump provides lubrication with a controlled flow rate, with the oil being pre-heated. A high temperature setting can be selected, as the walls of the test chamber can be heated to high temperatures. This along with a preheated lubricant oil, allows the PCC system to operate in realistic engine settings. Friction coefficients are recorded as a function of time by data collection in a computer. This data collection is continuous.

3.2. Application of Different Piston Rings Coatings

The experiment consisted of a centrifugally casted, pearlitic cast iron cylinder liner made of AT 182 Tarabusi Cie. with medium phosphorus content, with an inner diameter of 100 mm. The following three PR coatings were selected for the PR. The PR had a diameter of 90 mm:

- Chromated 126 = Chromium Plated
- CrN-TiN 1^o Group 127 = PVD multi-layer TiN-Ti-CrN-CrN coating
- NIPCO + Si₃N₄ 128 = Electroless nickel-phosphorus coating with dispersed silicon nitride (Si₃N₄) particles

The following test conditions were selected, and are summarized in Table 1 below:

Table 1. The testing conditions that were selected for each of the applied PR coatings.

Parameter	Condition Setting
Load	300 N
Contact Pressure	10 MPa
Test Time	5400 s
Frequency	50 Hz
Stroke	3 mm
Linear Speed	0.3 m/s
Temperature	200 °C
Lubricant	SAE 15W-40 ($\eta^{40C} = 107 \text{ mm}^2/\text{s}$, $\eta^{100C} = 14.6 \text{ mm}^2/\text{s}$, VI = 140, API SL/CF, ACEA A3/B3, A3/B4-08)

The oil temperature of 200 °C reflects operating conditions of engines at the piston ring or specifically at fired top dead center (FTDC), where adhesive failure (scuffing) frequently occurs [13].

4. Results

From Figure 5, this graph shows the COF of the different PR coatings over a duration of 5400 s.

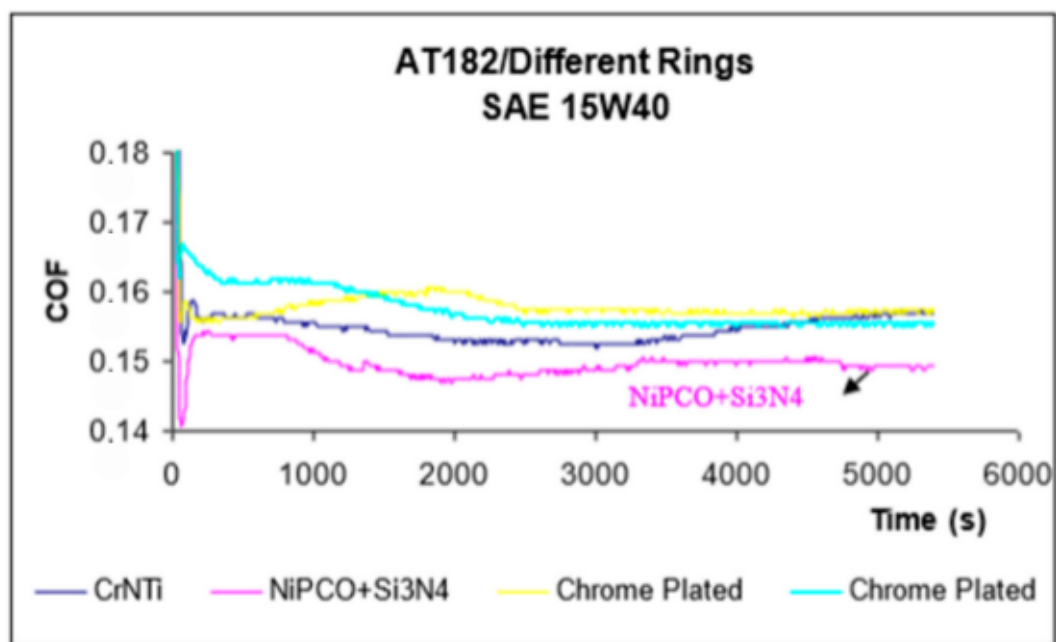


Figure 5. COF for different piston-ring coatings as a function of time.

The following graph (Figure 6) visually shows the PR mass loss:

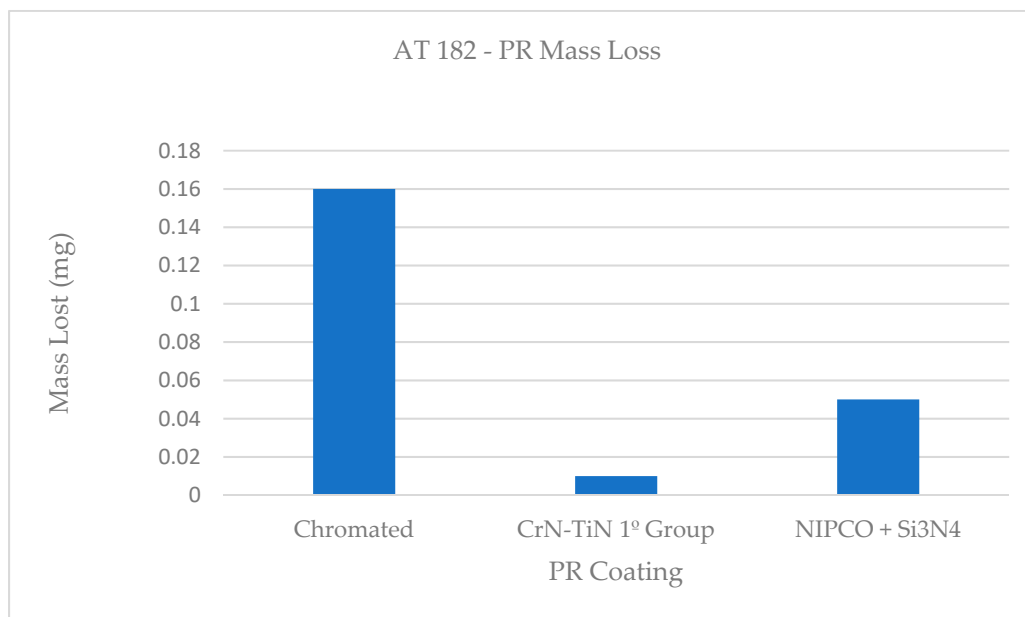


Figure 6. Loss of mass for each piston-ring coating.

The table (Table 2) below summarizes Figures 5 and 6:

Table 2. Summary of the Average COF and Ring Mass Lost for PR Coating.

Piston Ring	Average Coefficient of Friction	Ring Mass Lost (mg)
Chromium plated	0.158	0.16
CrN-TiN 1 st Group	0.156	0.01
NIPCO + Si ₃ N ₄	0.151	0.05

The chromium plated PR has the highest COF amongst the tested PR coatings at 0.158. Moreover, it also has the greatest loss of mass, 0.16 mg, of all three PRs tested. In this screening, the PVD multi-layer TiN-Ti-CrN-CrN and the electroless nickel-phosphorus coating with dispersed silicon nitride (Si₃N₄) particles coatings displayed a strong potential for wear reduction under scuffing conditions [14].

5. Discussion

From Figure 6, the results are as follows: the PR coating of CrN-TiN 1st Group lost the least amount of mass and the Chromium plated PR coating lost 16 times more in mass compared to the PR CrN-TiN 1st Group coating; and the PR NIPCO + Si₃N₄ coating lost 5 times more in mass compared to the PR CrN-TiN 1st Group coating. The PR coating of CrN-TiN 1st Group also has the lowest average COF. The oil bath of SAE 15W-40 provides the lubrication between the PR and the cylinder liner. This is done to help prevent the wear and yet each PR has experienced a loss of mass. As explained, over the Δx distance of the surface on the cylinder liner, there is a mixture of asperity points and well-lubricated points. Therefore, for each of the three PR tested, the oil film was penetrated allowing for wear to develop by the rubbing of the PR on the cylinder liner. However, it appears that the PR coating of CrN-TiN 1st Group had the least oil film penetration. This reasoning is valid, because of it having both the least loss of mass and the lowest COF among the tested PR coatings.

A very important physical property of a lubricant is its viscosity. Lubricant viscosity is dependent on the shear rate, temperature and contact pressure in the tribocontact [15]. The SAE 15W-40 is

a multigrade oil that experiences an elevated temperature of 200 °C and its viscosity can be calculated at any temperature through the Vogel Equation [15]:

$$\eta = \kappa \cdot e^{\left(\frac{\theta_1}{\theta_2 + T}\right)} \quad (1)$$

where η is the viscosity (mPa·s, where 1 mPa·s = 1 cP) at temperature T (°C) and κ (mPa·s), θ_1 (°C) and θ_2 (°C) are constants. Given that κ , θ_1 and θ_2 are constants, for a positive range of T values η is inversely proportional to T . The kinematic viscosity at 200 °C estimates to 3.18 mm²/s using this equation to model the viscometrics of the SAE 15W-40 lubricant during the experiment. In addition, multigrade oils are shear thinning fluids [16]. As such, the viscosity of the oil is affected by the shear rate, which in turn is dependent on the liner oscillation speed [7]. As the liner oscillation speed increases from static to a speed of 0.3 m/s, this may also contribute to the decrease in oil viscosity. Furthermore, the oil film thickness between the PR and cylinder liner increases with oil viscosity [17]; then conversely given the suspected decrease in viscosity, it hints to a decrease in the oil film layer, leading to a higher chance of wear on the PR.

The Stribeck curve described earlier is a model that can be applied to further explain the results. Due to the suspected decrease in oil viscosity, along with the normal load applied and linear oscillation speed remaining constant after its initial increase, the $\eta V/W$ factor probably falls. As this factor continues to fall, due to the eroding of the oil lubrication, the PR and cylinder liner rub together which causes wear on the PR, which also explains the loss of mass of the PRs but this model can simultaneously be used to explain Figure 5.

In Figure 5, for both the PR coating: CrN-TiN 1^o Group and Chromium plated, there is an initial increase in COF, with the chromium plated having a larger initial peak, then an immediate decrease in COF and finally increasing slightly in COF as time goes on. The Stribeck model explains that an initial increase in COF is followed by a decrease in COF till a minimum value. The cause for an immediate increase in COF can be attributed to the PCC contact going from no friction, as there is no linear velocity at time zero, to then a sudden increase in linear velocity. This in turn can cause the sudden increase in friction but is followed by an immediate decrease in COF. The reason for this being the high contact pressure in which the COF is generated in is also related to the oil viscosity [11]. The oil viscosity causes a reduction in friction in the initial stages, as there is enough lubrication to cause a decrease in COF. But as aforementioned, the linear oscillation is suspected to cause a decrease in oil lubrication making the COF rise eventually and cause wear to the PR. The anomaly to this trend is the NIPCO + Si₃N₄ coating, which has an initial decrease in COF and then a rise in COF as time moves on. A possible explanation is that over the contact distance, there were more pure hydrodynamic lubrication points, causing a reduction in COF. This did not last long as the COF rose quickly after, of which can be attributed to how the temperature and the linear oscillation speed change the lubrication pattern between the PR and cylinder liner. Ultimately, it did follow the trend of decreasing oil viscosity leading to PR mass loss and increased COF between the metals. The difference in the yellow and blue curve is due to running-in phenomena, which are mainly ruled by statistical events and can therefore produce significantly different COF-curves. These COF-curves depend on the many asperities that form through the actual contact between the piston ring and cylinder liner contact and how the lubricant reacts in these situations. Therefore, in this case the difference is *over-pronounced* by the scale of the graph. This is necessary to make the differences in COF between the coatings clearly visible and this all takes place within the range of the COF which is smaller than 0.02.

However, the PR coating of CrN-TiN 1^o Group still lost the least amount of mass. This can be attributed to the fact that CrN (Chromium nitrate) and TiN (Titanium Nitrate) are known for their good wear resistance [18]. More specifically, CrN and TiN are applied in racing car engines, where the PCC experiences greater piston speeds [19] due to the car travelling at high speeds during a race. For example, a Formula 1 engine V10-type experiences 37–48 m/s, whilst a regular European 2.0-L engine type Inline 4 has a maximum piston speed of up to 32 m/s [15]. The greater piston speeds that

a piston experiences in race cars indicate of the greater frictional force. Given that CrN and TiN are used for race car applications, it can be expected for them to exhibit good wear resistance, especially at a low speed of 0.3 m/s, as used during this experiment.

6. Conclusions

It was the hope of this study to help contribute more information to the field of optimizing the PR of the ICE. More specifically, the primary aim of this was to see the influence of the PR coating on the wear and friction on the PCC assembly. There is no proper correlation between wear and friction but if scuffing failure occurs, it can lead to material transfer and therefore wear. The coatings can show resistance to wear. Based on the experimental findings, it shows that a PR coating based from CrN and TiN has greater resistance to wear and friction. The wear and the COF experienced by the PR is a result of the complete tribosystem settings selected. This consists of the PR sliding against the cylinder liner, the elevated temperature, the surrounding atmosphere, load applied and the contact pressure. As such, the SRV offers these settings to help mimic the reality of what the PCC assembly faces during normal engine operations. Continued contributions to the understanding of the PCC is a critical area that will help improve efficiency and hopefully in a broader sense help automotive transports to be more effective with their fuel consumption.

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