A study into the effects of cylinder de-activation on the tribological performance of piston ring-liner conjunction

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A Study into the Effects of Cylinder De-Activation on the Tribological Performance of Piston Ring-Liner Conjunction

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With more demanding rules for greenhouse gas reduction, the need to manufacture more efficient vehicles provides the automotive industry with greater challenges to find novel methods to tackle this global problem. One such way in which this has been done is the application for cylinder de-activation (CDA) technology. In order to investigate the effects of CDA a 2-dimensional numerical model is used to predict the hydrodynamic effects in the piston – cylinder liner conjunction.

1. Introduction

Approximately 15-20% of losses in an IC engine are due to frictional power loss [1]. Of the frictional losses, 40-50% is contributed by the piston ring and piston skirt friction [2]. CDA is generally applied to large, multi-cylinder engines, and works by keeping the intake and exhaust valves closed through all cycles of the engine for a particular set of cylinders in the engine. The effects of CDA on big end bearing performance were investigated by Mohammadpour et al [3], finding that despite an improvement in overall fuel efficiency, there was a increased frictional power loss as a result of the reduced minimum lubricant film thickness leading to a higher degree of wear in components.

2. Methodology

Reynolds equation was discretized using a central finite difference method and point successive over-relaxation to obtain the pressure distribution [4]. The 2D Reynolds equation is [5];

$$\frac{\partial}{\partial x}\left[\frac{\rho h^3}{6\eta}\left(\frac{\partial p}{\partial x}\right) - \rho h U\right] + \frac{\partial}{\partial y}\left[\frac{\rho h^3}{6\eta}\left(\frac{\partial p}{\partial y}\right) - \rho h V\right] = 2\frac{\partial(\rho h)}{\partial t}$$

This form of Reynolds equation includes the effects of the flow in the direction of entraining motion, x and side-leakage from the contact, y. It also includes the effect of squeeze film action.

The solution requires calculation of the film shape, h, rheological equations for both density and viscosity variation with pressure under adiabatic conditions. For outlet boundary condition the Swift-Steiber (or Reynolds) conditions are used as they take into account the continuity of flow at the lubricant film rupture point. The inlet boundary is assumed to be fully flooded where the inlet pressure is the same as atmospheric pressure.

3. Results

A C-segment vehicle with a 4-cylinder 4-stroke gasoline engine is used in the current analysis [6], with an engine speed of 1500 r/min, corresponding to 35km/h on the NEDC when the car is driven in third gear. The results resemble (i) all active cylinders, (ii) an active cylinder in an engine with CDA and (iii) a deactivated cylinder in an engine with CDA.
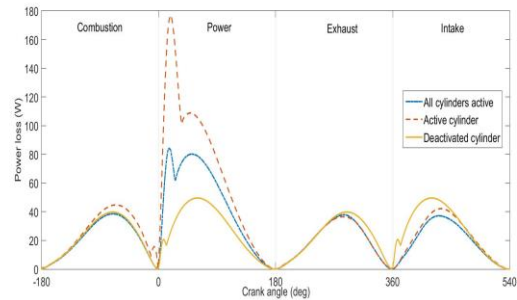

Figure 1: Cyclic Frictional losses [6]

Table 1: Average frictional power loss of the compression ring [6]

	Average Power Loss (W) for each ring
All cylinders active	26.44
Active Cylinder	33.87
Deactivated Cylinder	24.58

The analysis has shown that CDA has a significant effect on the frictional power losses (Figure 1), with the total frictional loss found to be 9.53% higher in an engine with CDA than one running under a normal operational load. The results show that the tribological performance of new automotive technologies should be accounted for.

4. Acknowledgement

5. References

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