

Piston Design for Optimizing Trade-off of Friction and NVH

Alastair Jay and Thomas Deighan
Ricardo UK Ltd.

Nozomu Kato and Kiyofumi Sato
Fuji Heavy Industries

ABSTRACT

Requirements for reducing powertrain NVH drives the selection of low piston skirt to liner clearances contradicting the requirement to maintain larger skirt clearances for minimizing engine friction. Whilst this clearance trade-off between low friction and low NVH is fundamental, piston design features have a significant effect on where the trade-off curve sits on the friction/NVH map. Design features can therefore be viewed not by either friction or NVH improvement measures but a shift in the friction-NVH trade off curve. Specifically, some piston design features which may be targeted at reducing friction can be viewed as either a friction benefit for similar NVH or an NVH improvement for similar friction levels.

The ability to realistically quantify the effect of the design changes on NVH is therefore critical to determining what design changes to recommend, the direction of the piston design being highly sensitive to the process by which the impact on NVH is assessed. Assessment of liner accelerations have frequently been used to quantify the effects of piston design features on NVH. However, this is not necessarily a good indicator of the perceivable NVH within the vehicle cabin, due to the variation in transfer path sensitivity and potentially dominating influence of other vibration sources across the speed and frequency range.

This paper demonstrates an analytical process for more realistically assessing the perceivable NVH due to the piston secondary dynamics whilst maintaining sensible analysis run-times. This process is applied in order to direct design improvements of a 2.5L gasoline engine piston to target reduced friction and NVH.

CITATION: Jay, A., Deighan, T., Kato, N., and Sato, K., "Piston Design for Optimizing Trade-off of Friction and NVH," *SAE Int. J. Passeng. Cars - Mech. Syst.* 9(3):2016, doi:10.4271/2016-01-1855.

INTRODUCTION

Engine manufacturers have been moving towards tighter installation clearances in an effort to reduce perceivable NVH levels in the vehicle cabin. Counter to this, the reduced clearances have the detrimental effect of going against continued efforts across the industry to increase engine efficiency and reduce whole engine friction. At a time where ever stricter legislation and customer expectations are coming to force, finding the correct balance for these two characteristics is a major driver for engine manufacturers.

A significant body of work has been conducted looking at how piston design characteristics affect skirt to liner friction and how the secondary motion of the piston affects piston noise characteristics such as piston slap and piston pin ticking [1 to 9]. In general, this work, whether physically tested or analysed, has used the acceleration as measured (or predicted) at the cylinder liner as an indication of how much noise is produced. Other work has been completed to try and characterize these noises and describe solutions to piston noise issues based on this characterization [11].

In this paper, a novel approach is used to look at how piston design parameters can affect NVH levels as predicted at the vehicle mounts and how these levels are traded off with friction.

A baseline analysis has been completed to produce a friction-NVH tradeoff curve, from which to assess potential design modifications. The analysis looked in detail at the dynamics of the piston to determine the mechanics producing the friction and vibration allowing identification of potential design changes which could affect where the piston lies on the tradeoff curve.

Following the baseline analysis, a sensitivity study is presented which details how the piston pin offset affects this position on the tradeoff curve. Modifications to the piston structure were also considered with a new design generated based on the latest benchmarking data and the latest published documentation [12, 13]. This was analysed using comparative installation clearances to quantify the benefit to friction and NVH. A comparison of the baseline and potential new design options are presented in terms of the friction-NVH tradeoff.

METHOD

Piston Secondary Loading

The piston secondary dynamic motion and resulting loading between the skirt and liner were calculated following the general process outlined in Figure 1.

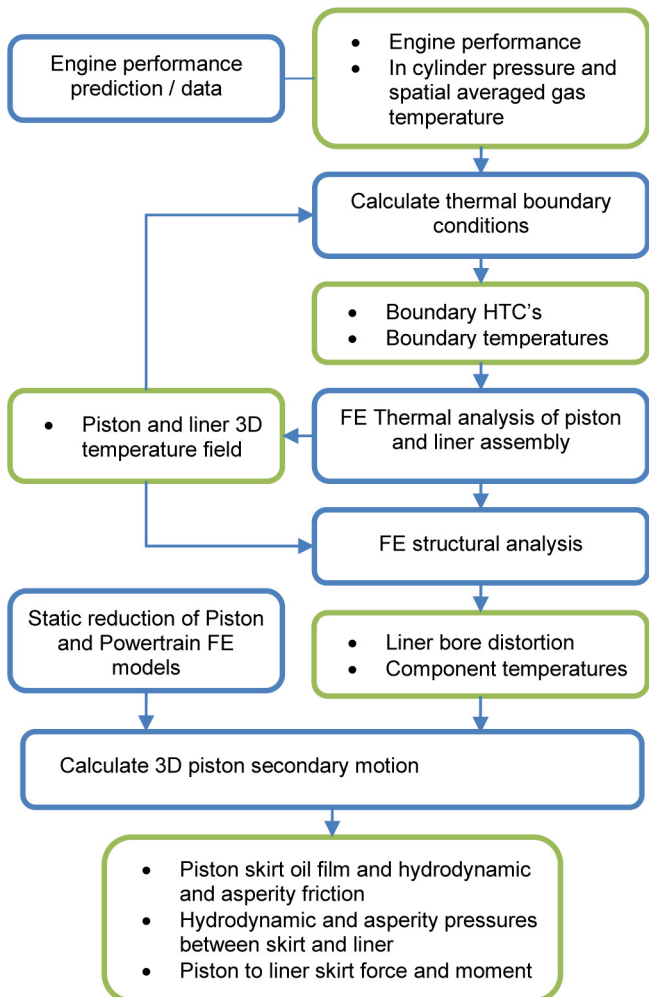


Figure 1. Process flow for calculation of piston secondary dynamics and loading

A similar process was used for previous work [1]. The following sections detail specific parts to this process critical for successful calculations of NVH.

Liner and Piston Deformation

The running clearance between the piston skirt and liner and the skirt profile shape is a dominant factor for determining the skirt friction and the predicted secondary vibration. The thermal deformations of the piston and liner under hot running conditions is therefore an important input to the calculations as they influence the general change in clearance from cold conditions and the detailed shape of the hot piston skirt profile.

Most critical for calculation of a sensible hot running clearance is the consistency of boundary conditions used for the temperature calculations of the piston and liner. For example, excessive heat input into the piston relative to the liner will result in over tight running clearance. As the piston temperatures influence the boundary conditions for the liner and vice versa, the boundary conditions were calculated following an iterative process based on consideration of incylinder gas conditions and the ring to liner and skirt to liner heat paths. Details of this calculation method is left to a later paper. The converged 3D temperature field is used in a FE structural analysis to calculate the piston and liner thermal deformations. This process was repeated for all running conditions considered for the analysis.

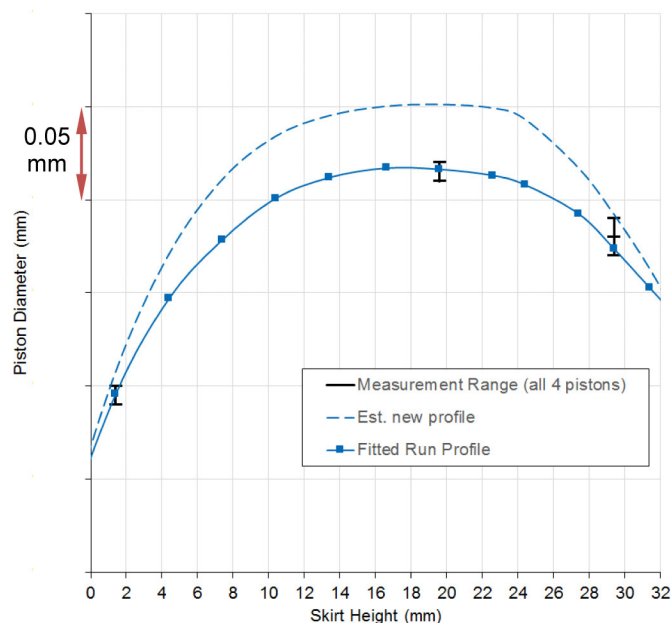


Figure 2. Piston diameter and skirt collapse

A second factor known to influence the skirt deformation and running clearance of the engine is 'skirt collapse', discussed by Mahle [11]. Mahle [11] show by measurement of the piston diameter before and after running a new piston that the skirt undergoes some permanent deformation. By utilizing specially made pistons with windows where the coating is removed the change in diameter due to coating wear and permanent skirt deformation can be separated. The measurements show a reduction in diameter of about 10 micron at the skirt height of profile maxima and much less (approx. 2micron) toward the top of the skirt. From the scale on the data presented in [11] the piston bore used for the measurements is approximately 70 mm. This deformation is significant compared to a corresponding cold radial running clearance of between 12 micron (tight for low NVH) and 35 micron (typical for good friction). The exact mechanism behind the skirt collapse does not seem to be well documented but is believed to be due to the creep resulting from the high temperature differences between the piston crown and skirt region. Measurements of skirt collapse were also made by Shibuya [10], stated to be due to creep, who noted the sensitivity to material thickness behind the lower ring groove. To account for skirt collapse the piston diameter was measured from a run engine (approx. 50 hours running). The measured data was used to estimate a modified cold skirt profile after running (Figure 2). Measurements from the same pistons prior to

running were not available but comparison to the design specification (with the piston known to be in specification) showed some level of skirt collapse must have occurred (Figure 2).

Structural Stiffness

Both the liner assembly (powertrain) and piston FE models can be represented within the analysis as a static or dynamic (CMS) reduction. For a static reduction the compliance of the structure is captured only. For a CMS reduction both the static compliance and component modes are represented.

Initial studies were performed by comparison of the secondary loading frequency content for compliant and dynamic reductions respectively. With the piston model it was found that whilst the compliance was of significant importance the effect on the results from using a CMS reduction were negligible until above 2.2 kHz. For the liner model it was also seen that frequency content was similar until above 2.2 kHz. It is also suggested however that if the liner dynamic response has a significant impact on the piston motion the coupling between the piston motion and powertrain dynamics is therefore important. The effect of the coincident interaction of the other pistons in the engine and reaction of secondary loads through to the cranktrain will therefore also be important. As a full co-simulation of piston dynamics for all 4 pistons with the powertrain and cranktrain was not within the scope of this project the analysis was conducted with compliant models, but with the above point kept in mind.

Secondary Dynamics

For this work PISDYN was used to model the dynamic motion of the piston. Within this program the elasticity of the liner and piston components are considered within an EHD calculation of the oil film between the piston skirt and liner. Asperity (boundary) contact between the surfaces under low film thickness conditions was calculated based on an algorithm from Greenwood and Tripp [14]. Further details on the methodology behind the calculation of the piston secondary dynamics is covered in [16 and 17]. Local oil temperatures between the skirt and liner interface are calculated from the adjacent component temperatures and used to determine the oil viscosity. A friction coefficient of 0.05 was used for the boundary lubrication based on the pressure rate of change of the limiting shear stress of the oil.

The lateral loading and moments (about the pin center) acting on the piston skirt are calculated by integration of the hydrodynamic and asperity contact pressures. The friction forces acting on the piston skirt and consequently the piston skirt FMEP are calculated from integration of the oil and boundary friction shear stresses.

Piston NVH Assessment

A number of previous works [1 to 6] have made assessments of NVH based on predicted liner accelerations. Kobayashi et al [1] showed a strong correlation between the subjective assessment in piston slap noise (under low speed, low load conditions) and the measured overall liner acceleration levels in dB. Prediction and relative comparison of liner accelerations could then be related back to the

subjective rating. Whilst the assessment of liner accelerations is a useful measure of the subjective change in piston noise it has the following limitations,

- Prediction of sensible liner accelerations normally requires a dynamic liner assembly model (CMS) as twice differentiating the calculated displacements from analysis with a compliant model results in numerical noise
- The frequency dependent transfer path of the piston loading to the engine mounts and radiated noise is not captured
- The contribution to mount vibration (or radiated noise) of the piston secondary loading, relative to other sources (eg. crankshaft dynamics, combustion, valvetrain dynamics etc...) is not evident

The work presented in this paper was part of a larger study [15] assessing the influence on NVH from cranktrain, valvetrain and piston and it was therefore required to assess the mount vibration due to the piston vibration in the context of the other loading sources. The primary piston side force loads were calculated from the slider crank kinematics within a dynamic simulation of the crankshaft, cosimulated with the powertrain dynamic response. Details of the cosimulation with additional dynamic models and NVH post processing is covered in detail in [15]. In conjunction with this work the process outlined in Figure 3 was followed to assess the mount vibration and radiated noise from the piston.

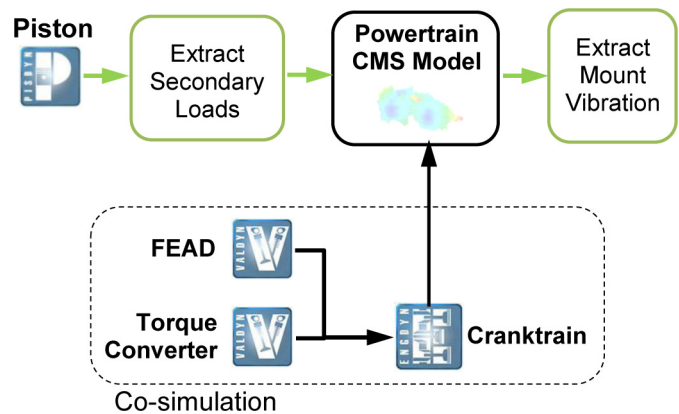


Figure 3. Process for assessment of contribution of piston secondary dynamic on powertrain NVH

Extraction of Secondary Loading

The total loads acting between the skirt and liner on the thrust and anti-thrust sides were calculated separately from the pressures obtained from the piston dynamic simulation. The secondary loading content was then extracted by subtraction of the primary loads from calculation of the slider crank kinematics. Figure 4 shows an example of the total and extracted secondary loads across the engine cycle.

The secondary loading is (as would be expected) indicative of the liner accelerations and therefore allows direct assessment of the frequency content of the vibration at the liner without the requirement for a dynamic liner model. Although the frequency content of the

secondary loading was assessed directly for some design comparisons within this work, the main output used for quantification of the piston NVH was the vibration at the engine mounts.

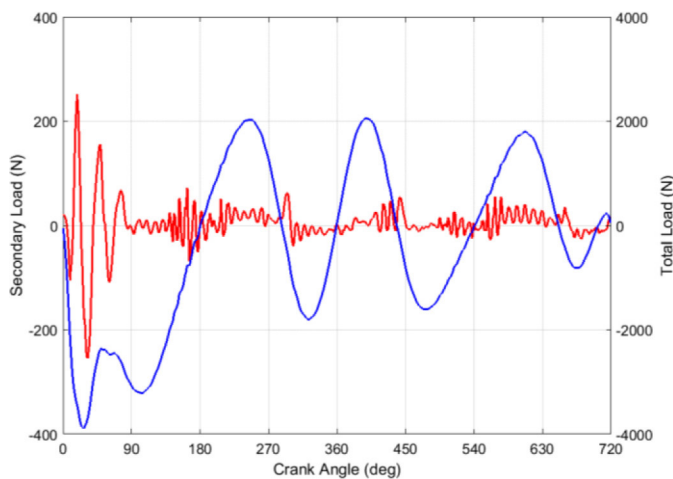


Figure 4. Example of extracted resultant secondary loading, variation over the engine cycle

Mount Vibration Assessment

The extracted secondary loading was applied to the powertrain model CMS reduction used in [15], in addition to other loading from combustion and cranktrain dynamics. Figures 5a and 5b show the predicted mount vibration at full load under hot and cold start conditions. Comparisons were made across an 81.5 micron range of cold radial clearances.

- Baseline (for targeting minimal piston NVH)
- Typical lower bound
- Typical upper bound

The vibration is also compared to the predictions made without the piston secondary loading applied.

The results shown in Figure 5a show that due to the tight hot running clearance contribution of the piston vibration to the mount vibration is negligible across the speed range and up to the 2kHz octave band. As the specified cold clearance is increased across a typical range (for lower friction) of 47 to 94 micron the contribution progressively increases, most significantly at higher frequency and lower speed. Similar trends were also seen at part load and when assessing radiated noise from the block.

At higher speed the contribution to the mount vibration from the piston remains relatively small even with the larger clearances. Although the piston secondary dynamics and loading are greater at higher speed the level of vibration at the mounts is dominated by other sources such as the crankshaft dynamics and combustion loading.

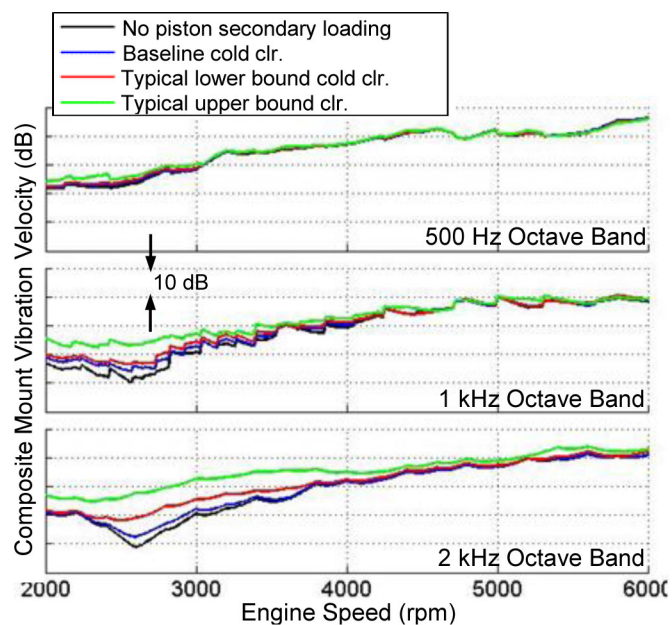


Figure 5a. Composite mount vibration at Full Load hot conditions

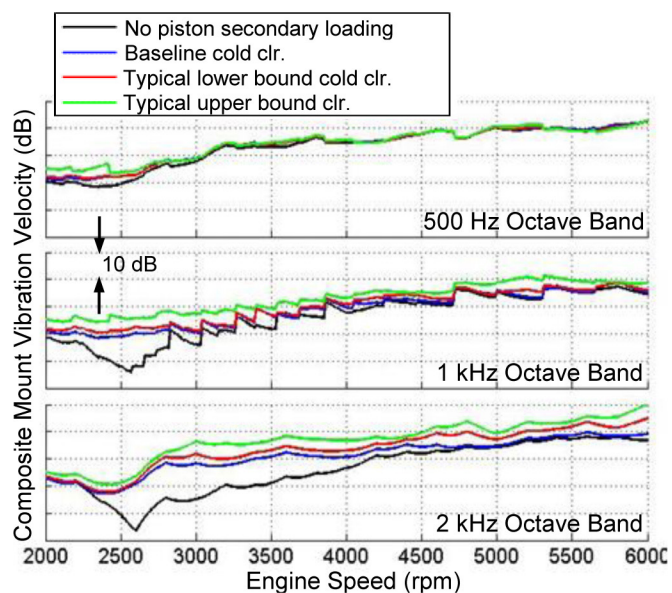


Figure 5b. Composite mount vibration at Full Load cold conditions

The results in Figure 5b show that under cold conditions the contribution to mount vibration from the piston is significantly increased. The engine does not operate under 'cold' conditions for long periods but provides an upper bound to the level of vibration due to piston secondary motion during warm-up. With the baseline clearance level the vibration is still predominantly seen at low speed and high frequency, whereas at the larger clearances it is more significant across the speed range.

The increased relative contribution of the piston secondary loading to the mount vibration under cold conditions occurs for two main reasons.

1. The piston secondary motion and resulting loading is significantly increased under the increased piston to liner clearance without the relative thermal expansion of the piston within the liner