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MODELING OF PISTON SLAPPING MOTION

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ABSTRACT

The aim of this work is to develop various analytical models for studying the piston slapping motion. The results of various methods are compared to the experimental results using block vibrations measurements on a diesel engine test rig. Both results correlate, validating the considered theoretical methods.

Keywords: piston motion, slap

1. INTRODUCTION

Piston slap is a major source of noise in automotive industry, which is a cause of concern for many car makers in order to devise methods for compilation with various emission norms, as well as to meet customer satisfaction. With many developments in this industry, the problem of NVH has become more significant, particularly in case of diesel engines. The decade of 60s saw the first attempt at governmental noise regulations that directed its efforts toward the automotive industry [1]. During the decade of 70s, new methods were developed by De Jong to study the piston slap motion, with focus on path for noise transmission [2]. Later, Ohta et al. presented an analytical model, taking into account block vibrations [3]. Later on, more work was done by Kamp and Spermann to evaluate and improve piston-related noise in engines [4]. Isuzu has also presented a mathematical model to study the piston secondary motion, taking into account hydrodynamic behavior of lubrication oil.

2. THEORETICAL BACKGROUND

Piston secondary motion is due to striking of piston, with major thrust and anti-thrust sides of liner. Hence, it is necessary to develop an analytical model to study the piston dynamics motion along with the engine block vibrations, considering the piston transverse motion, the tilt angle and various contact forces. Many models have been discussed here to understand this motion. First, a point mass model of the piston slap has been discussed, which simulates the secondary motion, as shown in Fig. 1.

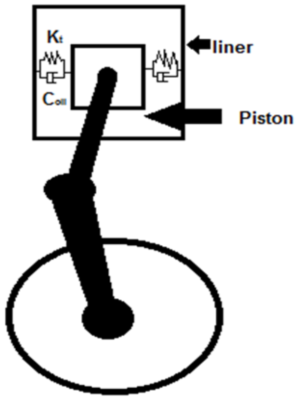

Fig. 1. Point mass analogy model

The governing equations for this system can be expressed in terms of skirt-liner gap (e), mass matrix (m) and damping matrix (C) as [5]:

$$mx''+Cx'+K(x\pm e)=F_{applied} \quad (1)$$

This equation can be solved using MATLAB23S command for which transformation from second order differential equation to first order is needed. Hence, transformations were done as under these equations:

$$x=y_1 \quad (2)$$

$$x'=y'_1= y_2 \quad (3)$$

$$y'_1= y_2 \quad (4)$$

$$y'_2=\frac{-K}{m}\,y_1+\frac{-C}{m}\,y_2\pm\frac{-K}{m}+F_{applied} \quad (5)$$

Arranging in matrix form we have:

$$\begin{bmatrix} y'_1 \\ y'_2 \end{bmatrix}=\begin{bmatrix} 0 & 1 \\ \frac{-K}{m} & \frac{C}{m} \end{bmatrix}\begin{bmatrix} y_1 \\ y_2 \end{bmatrix}+\begin{bmatrix} 0 \\ \pm\frac{K}{m}+\text{Fapplied} \end{bmatrix} \quad (6)$$

The required dynamic features including dynamic mass (m), dynamic stiffness (k), dynamic damping (c) and first resonant frequency (ω_a) of piston-liner system were calculated by modal analysis as seen in Table 1. The output is piston displacement which is discussed later.

Table 1 –Dynamic Specifications of engine

Test Case	Parameter	Value
	Piston Parameter	Liner Parameter
1	ω_a=100 Hz	ω_a=39 Hz
	c=109330 kg/s	c=42884 kg/s
	k=174 kg/s	k=175 kg/s^2
	m=174 kg	m=175 kg
2	ω_a=100 Hz	ω_a=39 Hz
	c=109330 kg/s	c=42884 kg/s
	k=174 kg/s^2	k=175 kg/s^2
	m=174 kg	m=175 kg
3	ω_a=158 Hz	ω_a=63Hz
	c=172750 kg/s	c=69669 kg/ s^2
	k=174 kg/s^2	k=175 kg/ s^2
	m=174 kg	m=176 kg
4	ω_a=158 Hz	ω_a=63Hz
	c=172750 kg/s	c=69669 kg/ s^2
	k=174 kg/s^2	k=175 kg/ s^2
	m=174 kg	m=176 kg
5	ω_a=158 Hz	ω_a=63 Hz
	c=172750 kg/s	c=69669 kg/ s^2
	k=174 kg/s^2	k=175 kg/ s^2
	m=174 kg	m=176 kg

Next model considers bar analogy for cylinder liner [6].The motion equation can described in terms of impact forces, $\sum \boldsymbol{F_s}$, as:

$$mx''_p= F_{applied} + \sum \boldsymbol{F_s} \quad (7)$$

Vibrations in bar can be analysed by following equation in terms of Young modulus of piston (E), piston crown area (A) and piston velocity (u):

$$EA\frac{\boldsymbol{du^2}}{\boldsymbol{dx^2}} = \rho u\frac{\boldsymbol{du^2}}{\boldsymbol{dt^2}} \quad (8)$$

This relation can be solved by separation of variables to get natural modal frequencies (ϖ_n) and mode shapes (ϕ_n) of oscillations as:

$$\varpi_n = \frac{(2n-1)\pi x}{l} \quad (9)$$

$$\phi_n = B\,Sin\left(\frac{(2n-1)\pi x}{l}\right) \quad (10)$$

The magnitude of longitudal vibrations of bar g(t) may be represented in form of following equation:

$$g''(t) + 2\,\varpi\, g'(t) + \varpi^2 g(t) = f_i \quad (11)$$

where

$$f_i = \frac{+\sqrt{\frac{2}{ml}}}{-\sqrt{\frac{2}{ml}}}\ \sum F_s \quad (12)$$

The governing equations for this system can be expressed in terms as:

$$x_p' = y'_1 = y_2 \quad (13)$$

$$y'_2 = -\varpi\, y'_1 - 2\xi\, y_1 + \sqrt{\frac{2}{ml}}\sum F_s \quad (14)$$

These equations were solved using Runger Kutta method in MATLAB resultng in plots of transverse motion of bar which is discussed later.

Next analytical model considers vibrations of cylinder block along with piston motion [7]. The equations of this system representing net forces (f_x) and moments (M_x) has been derived in terms of piston mass (m_p), connecting rod mass (m_r), tilt angle of piston (θ_p), connecting rod tilt angle (β), piston assembly moment of inertia (I_P), horizontal offset of centre of mass from piston pin (L_x) and vertical offset of centre of mass from piston pin (L_y) [8].

$$f_x = (m_p + m_r)\, x''_p + m_p(L_y - L_x \mathrm{Tan}\,\beta)\theta''_p \quad (15)$$

$$M_x = (I_p + m_p(L^2_x + L^2_y))\,\theta''_p + m_p L_y\, x''_p \quad (16)$$

These equations can be solved by MATLAB Runger kutta method giving tilt angle as well as transverse displacement of piston.

3. EXPERIMENTAL SETUP

Experiments were conducted on a dual cylinder lombardini LDW442CRS common rail direct injection test rig having specifications as presented in Table 1.

Table 2 -Specifications of engine

Bore	68 mm
Stroke	60.6mm
Displacement	440 cc
Rated Power	8.5kW @2000 RPM
Rated Torque	25N-m @4400 RPM
Compression Ratio	20:1

A fully opened electronic control unit connected to computer was used to manage the injection system with aim to control operational parameters. The engine was couled with an a synchronous motor of SIEMENS 1PH7 thus allowing to control speed and load. A Bruel and Kjaer free field microphone of 4939 type with a 2670 type preamplifier was used to obtain acoustic data having specifications as seen from Table 5. This engine test rig has a piezoelectric type Kistler 6056A pressure transducer (specifications as presented in Table 5) for in-cylinder pressure measurements and an optical crank angle encoder of AVL 364C for detection of TDC position as well as engine speed.

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Table 3 -Specifications of tests

Case	RPM	Load	P_{rail}
1	2000	80%	508
2	2000	100%	714
3	3000	80%	515
4	3000	100%	710
5 (motored)	3000	100%	-

Table 4-Specifications of Fuel Injection

Case	Q_{main}	Q_{Pre}	SOI_{main}	SOI_{Pre}
1	6.3	1	5.09°	19.9°
2	13.9	1	6.29°	14.6°
3	6.6	1	5.68°	22.5°
4	13.8	1	6.2°	16.5°
5 (motored)	-	-	-	-

The given system can do maximum 2 injections per cycle. All sigals were simulatneously acquired by NI boards of 6110 type (for analog type) & 6533 type (for optical encoder signals) using LabVIEW 10 software. During the tests, the sampling ratye was varied in order to guarantee a resolution of 0.25° CAD. The engine was operated at speeds of 2000 RPM and 3000 RPM under various loaded and motored conditions, as seen from Tables 3 and 4.

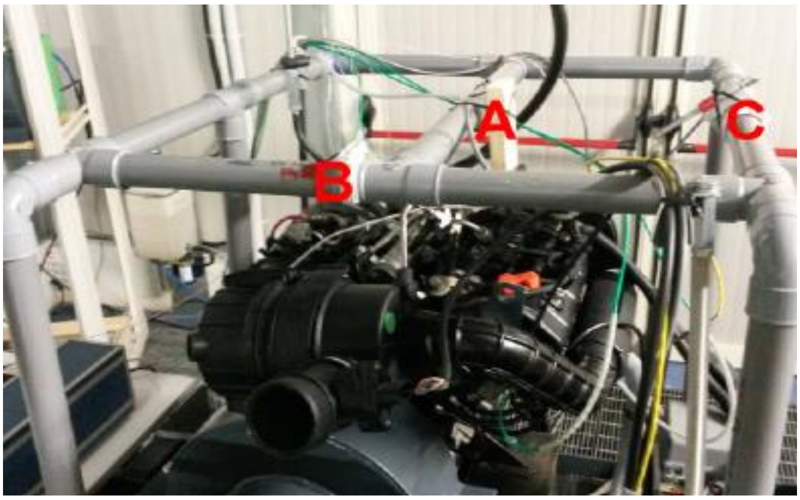

Fig. 2. Test Rig Showing Micro phone

Table 5. Specifications of microphone transducer

Range	4 mV/Pascal
Sensitivity	4-1000 kHz

Table 6. Specifications of pressure transducer

Range	0-250 Bar
Sensitivity	20 pC/Bar
Resonance Frequency	160 kHz

Table 7. Specifications of Accelerometer transducer

Range	1000 g
Sensitivity	3 pC/g
Resonance Frequency	90 kHz

Block vibrations were measured using ENDVECO 7240 C accelerometer, mounted on all three positions (A, B, C). Main features of this accelerometer are presented in table no 7.

4. RESULTS AND DISCUSSIONS

Figures 3 and 4 show the piston motion calculated by point mass analogy. It is evident from the figures that point mass strikes bore penetrating inside, where force gets balanced. The direction of force changes as the point mass rebounds against the bore and moves to other side.

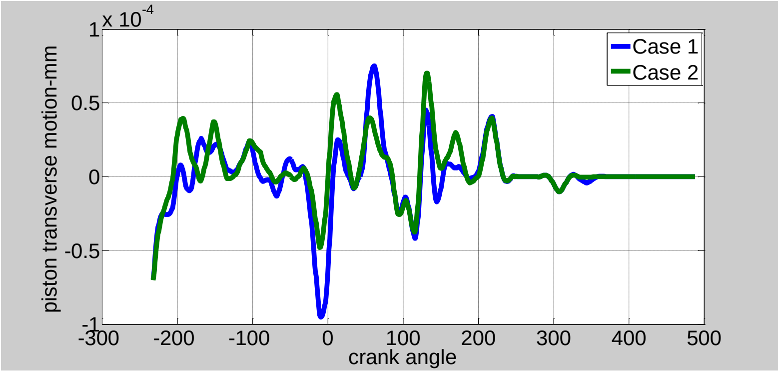

Fig. 3. Piston motion (Point mass analogy -2000 RPM)

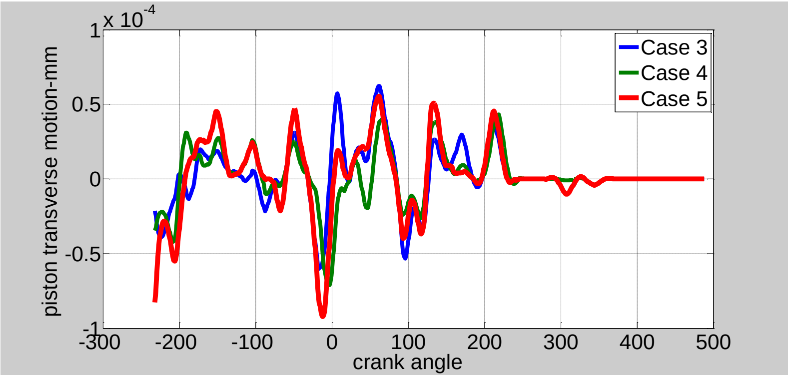

Fig. 4. Piston Motion (Point mass analogy-3000 RPM)

The results obtained from bar analogy are shown in Figs 5 and 6. The point mass starts its motion from center position, between right and left bar. It moves towards positive direction hitting the bar, rebounding and then trancelling towards negative side. As point mass separates from the bar, the oscillations of bar take place, continuing until the next contact.

Results obtained by analysis of third model are seen in Figures 7-11. Both actual and numerical simulation showed as good correlation. The piston moves from the bottom dead center to the top dead center (TDC), rotating clock wise, near TDC position. At the crank angle position 15°, after top dead center position, the piston tilts in counter clock wise direction. This is the instance of pison slapping motion. Later, in the compression stroke, the piston rebounds and travels towards major thrust side, striking the bore and causing liner oscillations. This phenomenon repeats again and again, throught out the engine cycles.

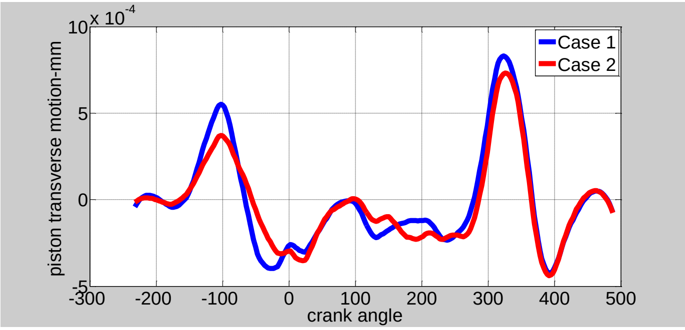

Fig. 5. Piston motion (Bar analogy - 2000 RPM)

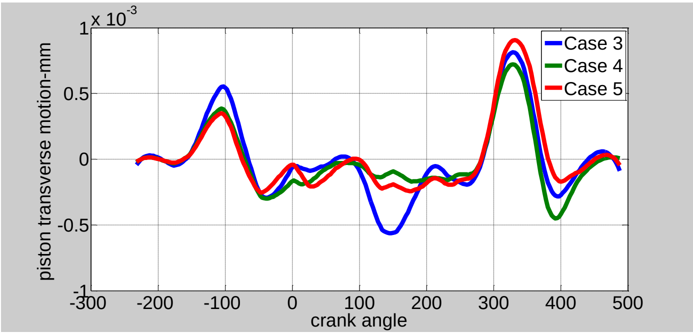

Fig. 6. Piston motion (Bar analogy - 3000 RPM)

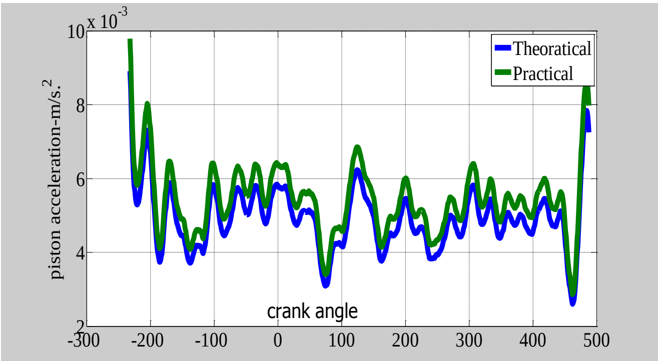

Fig. 7. Piston acceleration (Case1)

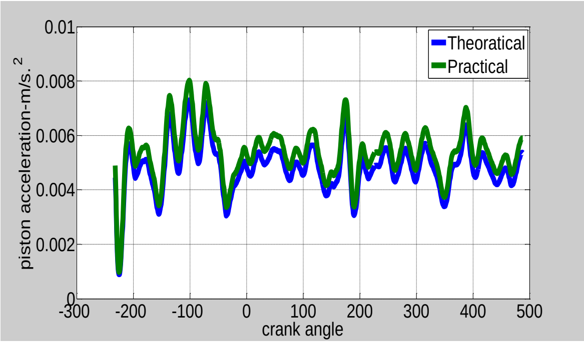

Figure 8-Piston acceleration (Case2)

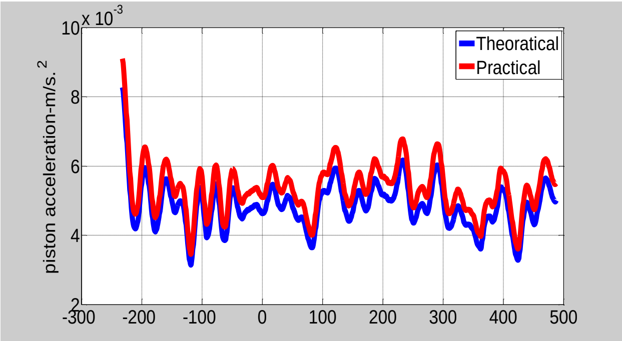

Fig. 9. Piston acceleration (Case3)

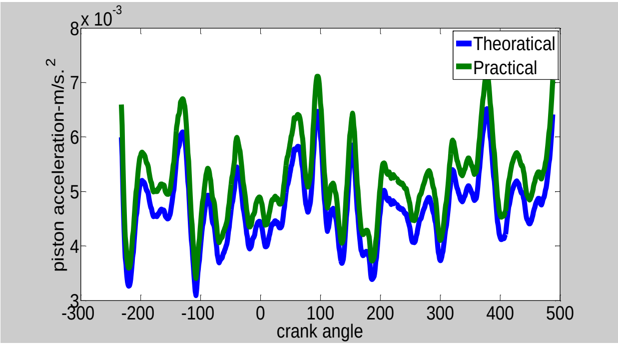

Fig. 10. Piston acceleration (Case4)

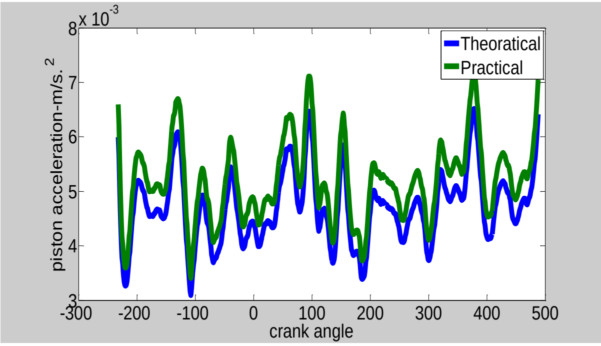

Fig. 11. Piston acceleration (Case5)

5. CONCLUSION

In this study, mathematical models were analysed for understanding the piston slap in a diesel engine. Several experimental tests were done to verify these models. A high degree of correlation was found between the theoretical and experimental results. These models could be applied on both gasoline and diesel engines.

NOMENCLATURE

Q_{pre} - Amount of fuel injected during pre-injection Period (mm^3per stoke)
Q_{main} - Amount of fuel injected during main-injection Period (mm^3per stoke)
SOI_{pre} -Angle of start of pre-injection period (degrees before Top Dead Center)
SOI_{main} - Angle of start of main-injection period (degrees before Top Dead Center)
P rail - Injection Pressure of fuel inside cylinder
BTDC - Before Top Dead Center

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