DYNAMIC MODELLING OF PISTON THE MOTION IN COMBUSTION ENGINES

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Resume

The presented work discusses a methodology for analysis of noise emissions from a diesel engine. A numerical model of the piston motion, analyzing its lateral, reciprocating and rotation motion, has been presented in order to investigate the lateral motion of the piston skirt assembly and resulting vibrations induced as a result of these motions in the engine block. Various parameters of modal analysis were obtained using the mobility analysis. The presented methodology was validated by data obtained from a diesel engine test set up. The predicted results matched well with those of measured data, hence validating the presented scheme.

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1 Introduction

In combustion engines a lateral space is present between the skirt and a cylinder liner that gives a motion freedom in lateral direction during the engine operation [1]. The existence of this gap puts a limit on magnitude of piston motion [2]. The piston assembly contributes to about 30-40% of mechanical losses and hence its design is a major concern for automotive engineers [3-4]. The piston thrusts liner to other side due to changing in direction of side thrust force due to motion of a connecting rod [5-6].

A dynamic model of the crank slider mechanism has been presented by Flores et al. [2]. The existence of lateral gap makes the system nonlinear and chaotic in nature. The reaction force between the liner and a skirt also plays an important role in dynamics of motion. As the coefficient of restitution decreases, the motion transforms from bouncing to a periodic one [7-8].

McFadden and Turnbul analyzed effects of combustion gas pressure on primary motion of a piston [9]. A two degree of freedom system has been analyzed showing a correlation between the piston slap and resulting vibrations [10-16]. Various parameters affecting piston motion has been considered, which includes location of a center of gravity [17], profile of a skirt [18-19], effects of inertial forces [20-21], frictional forces [22] and lubricating oil [23]. Mounted accelerometers on the block surface were used to simulate the piston's secondary motion [5].

2 Piston assembly model

The secondary motion of a skirt for the case of a 240 cc engine was modeled as depicted in Figure 1. The piston was considered as a point mass of 0.363 kg (m_p) and inertia (I_p) of 7.8540X10^{-9} kg-m^2 having two degree of freedom in motion (X_p ,θ). The cylinder block was considered as a lumped mass of 48.5 kg (m_b) with a single degree of freedom X_b, as shown in equation (1).

The nominal clearance of 0.5 mm allows the piston assembly to move in the lateral direction, as well as to rotate about the piston pin. The clearance between the skirt and a liner Xc was modeled as a mechanical stop in lateral direction.

For condition of no impact, (X_p -X_b=X_c) the motion was governed by Equation (1).

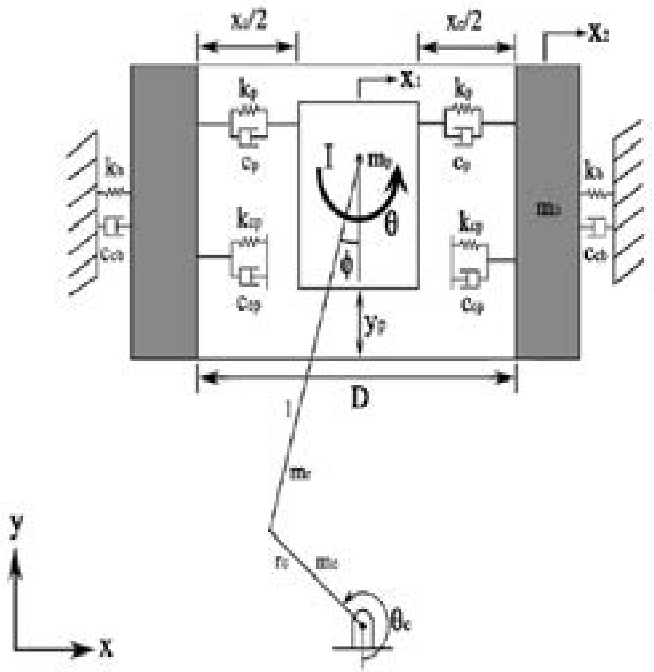

Figure 1 *Numerical model of the piston motion*

$$\begin{pmatrix} m_p & 0 & 0 \\ 0 & m_b & 0 \\ 0 & 0 & I_p \end{pmatrix}\begin{pmatrix} X_p'' \\ X_b'' \\ \theta'' \end{pmatrix} + \begin{pmatrix} C_p & -C_p & 0 \\ -C_p & C_p + C_b & 0 \\ 0 & 0 & C_\theta \end{pmatrix}\begin{pmatrix} X_p' \\ X_b' \\ \theta' \end{pmatrix} + \begin{pmatrix} k_p & -k_b & 0 \\ -k_p & k_p + k_b & 0 \\ 0 & 0 & k_\theta \end{pmatrix}\begin{pmatrix} X_p \\ X_b \\ \theta \end{pmatrix} = \begin{pmatrix} F_x \\ 0 \\ M_z \end{pmatrix}. \quad (1)$$

3 Piston side thrust force

The major issue, affecting this lateral motion of a skirt, is the side thrust force (Fx) imparted to skirt by a connecting rod, as shown in Figure 2.

The frictional forces act between the piston skirt and a cylinder liner (F_f) as well as between the rings and a liner (F_{fr}). The force exerted by the connecting rod on piston pin was resolved along X (F_{rodX}), as well as Y axis (F_{rodY}). The side thrust force (F_x) takes into consideration both inertial forces as well as gas forces (F_g) [21-24].

$$F_x = [F_g - m_p r\omega^2[\cos(\theta)+K\cos(2\theta)]]\lambda, \quad (2)$$

where: m_p - mass of piston,
θ - crank angle,
ω - angular speed,
r -crank radius,
K - crank radius-connecting rod length ratio,
$\lambda = K\cos(\theta)/\sqrt{(1+\sin(\theta)]^2)}$.

4 Piston frictional force

Various friction forces play a predominant role in the total mechanical loss of an engine [25-26]. According to Zweiri et al. [27], frictional force between the rings and a liner can be obtained from the product of elastic tension and the coefficient of frictional force. As the speed of engine increases, the coefficient of friction decreases gradually until reaching the minimum at the mid stroke. The frictional forces between the liner and a skirt (F_f) and piston rings and liner (F_{fr}) may be expressed in terms of the sliding velocity of a piston (V), nominal clearance (h), lubricating oil viscosity(μ), number of piston rings (n) and the shear area of a contact (A_s) as [27-57]:

$$F_f = \mu V A_{s1}/h, \quad (3)$$

$$F_{fr} = n\mu V A_{s2}/h, \quad (4)$$

where A_{s1} is the shear contact area between the liner and a skirt and A_{s2} is the shear contact area between the liner and rings.

5 Mobility parameter determination

The mobility may be defined as the ratio of velocity response V(Jω) of a structure to exciting force F(Jω) acting on a structure [5]:

$$M(J\omega) = V(J\omega)/F(J\omega), \quad (5)$$

$$M(J\omega) = -J\omega((K-M\omega^2)+JC\omega)/M\omega^2(K+JC\omega), \quad (6)$$

In the frequency range below the first anti resonance frequency value (ω_a = K/m), the point mobility equation can be approximated as [5, 31-32]:

$$M(J\omega) = -J/m\omega_a. \quad (7)$$

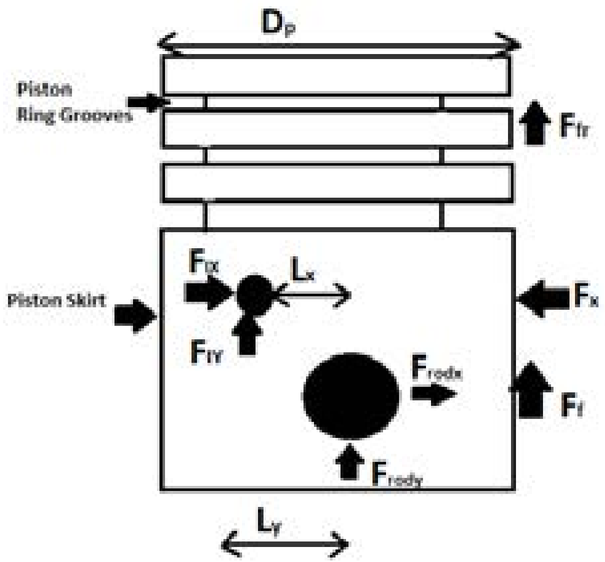

Figure 2 *Force diagram of the piston skirt assembly*

Above the anti resonance frequency, the point mobility can be written as :

$$M(J\omega) = -J\omega_a/K. \tag{8}$$

6 Experimental setup

Tests were done on a single cylinder HARTZ engine having specifications as presented in Table 1.

The in-cylinder pressure was monitored by an AVL transducer, having specifications shown in Table 2. Block vibrations were measured by means of an Endveco7240C type Mono axial accelerometer, having features accelerometer are presented in Table 3.

Various engine testing speeds in rpm (Revolutions per minute)-(2000 rpm and 3000 rpm) and load values (80% and 100%) were chosen with an aim to cover complete engine operational conditions. The data recorded during each test was under steady state conditions as seen in Table 4.

Figure 3 shows the general layout of the test rig with placement of various sensors.

7 Results and discussions

Figures 4 and 5 depict variations of the piston side thrust force. This force changes its direction five times in a complete engine cycle indicating five possible instances of lateral contact of the skirt with a liner.

COMSOL 7 multi physics software was used to

Table 1 *Engine specifications*

Type	Diesel Engine
Make	HARTZ
Number of cylinders	1
Bore	69 mm
Stroke	65 mm
Displacement	0.243 liter
Compression	22:1
Maximum power	3.5kW @ 4400 rpm
Maximum torque	10N-m @ 2000 rpm

Table 2 *Pressure transducer specifications*

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

Table 3 *Accelerometer specifications*

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

Table 4 *Testing specifications*

Case	rpm	Load	$P_{injection}$ (Bar)
1	2000	80%	716
2	2000	100%	692
3	3000	80%	814
4	3000	100%	612
5	3000	-	512

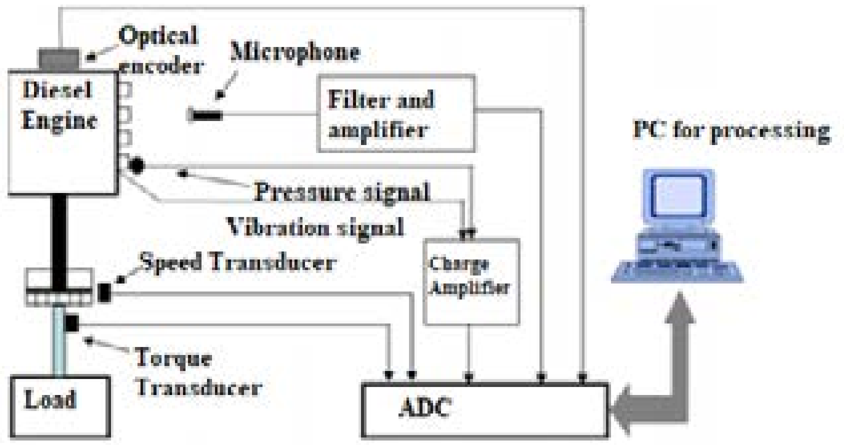

Figure 3 *Experimental setup*

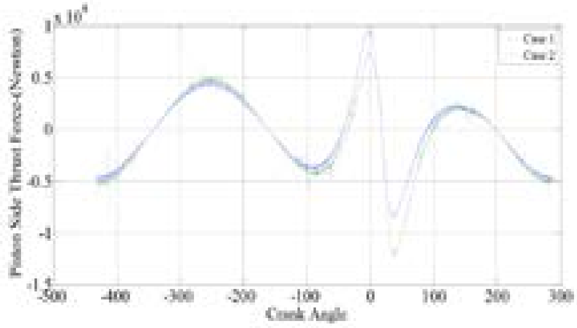

Figure 4 *Variations of the piston side thrust force (2000 rpm)*

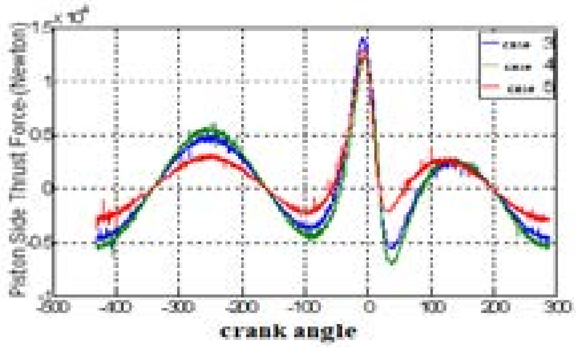

Figure 5 *Variations of the piston side thrust force (3000 rpm)*

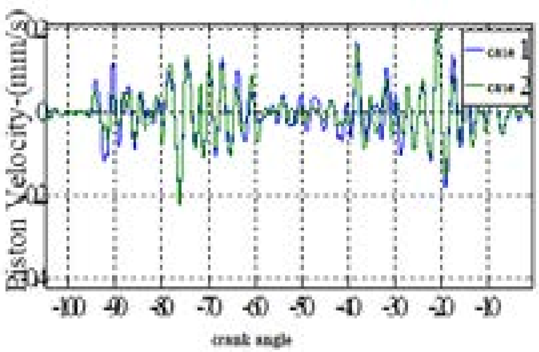

Figure 6 *Variations of the piston velocity (2000 rpm)*

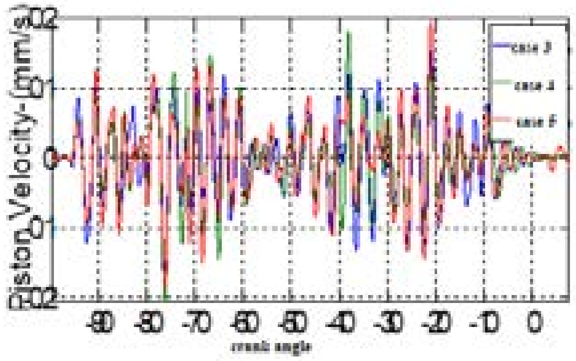

Figure 7 *Variations of the piston velocity (3000 rpm)*

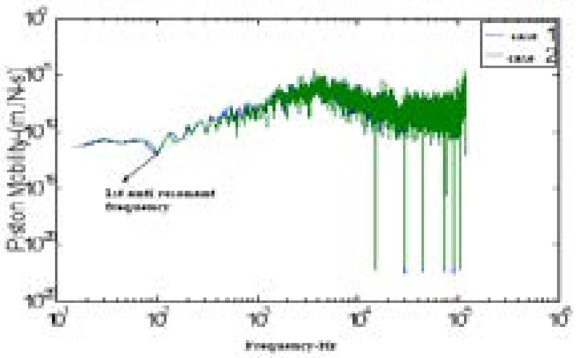

Figure 8 *Variations of the piston mobility (2000 rpm)*

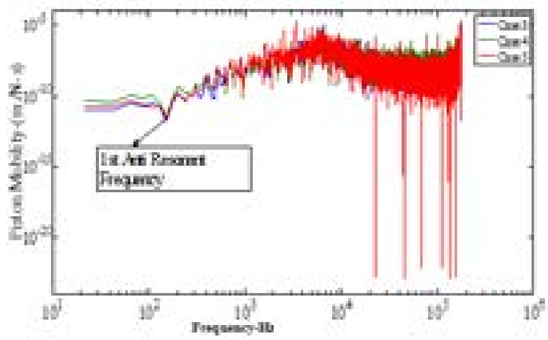

Figure 9 *Variations of the piston mobility (3000 rpm)*

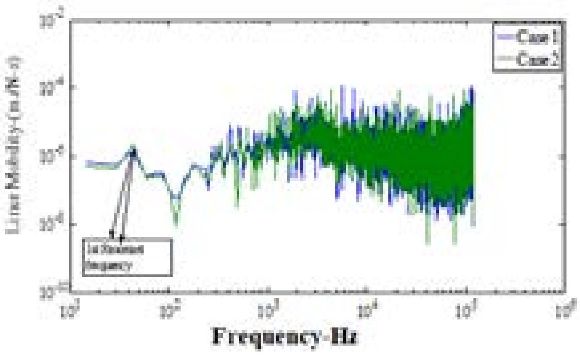

Figure 10 *Variations of the piston mobility (2000 rpm)*

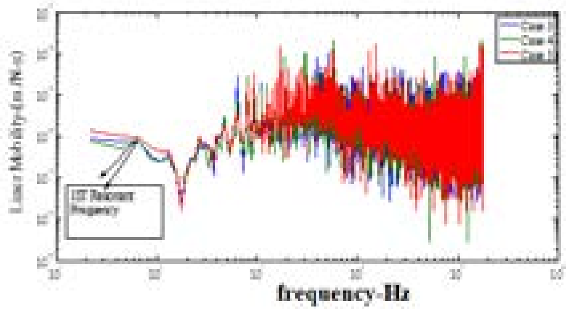

Figure 11 *Variations of the piston mobility (3000 rpm)*

Table 5 *Dynamic features of a system*

Test case	Piston parameter	Liner parameter
1	ω_a 100 Hz	ω_a 39 Hz
	C_p 109330 (kg/s)	C_b 42884 (kg/s)
	K_p 174 (kg/s^2)	K_b 175 (kg/s^2)
	m_p 174 (kg)	m_b 175 (kg)
2	ω_a 100 Hz	ω_a 39 Hz
	C_p 109330 (kg/s)	C_b 109330 (kg/s)
	K_p 174 (kg/s^2)	K_b 174 (kg/s^2)
	m_p 174 (kg)	m_b 174 (kg)
3	ω_a 158 Hz	ω_a 63 Hz
	C_p 172750 (kg/s)	C_b 69669 (kg/s)
	K_p 174 (kg/s^2)	K_b 175 (kg/s^2)
	m_p 174 (kg)	m_b 176 (kg)
4	ω_a 158 Hz	ω_a 63 Hz
	C_p 109330 (kg/s)	C_b 109330 (kg/s)
	K_p 174 (kg/s^2)	K_b 174 (kg/s^2)
	m_p 174 (kg)	m_b 174 (kg)
5	ω_a 158 Hz	ω_a 63 Hz
	C_p 172750 (kg/s)	C_b 69669 (kg/s)
	K_p 174 (kg/s^2)	K_b 175 (kg/s^2)
	m_p 174 (kg)	m_b 176 (kg)

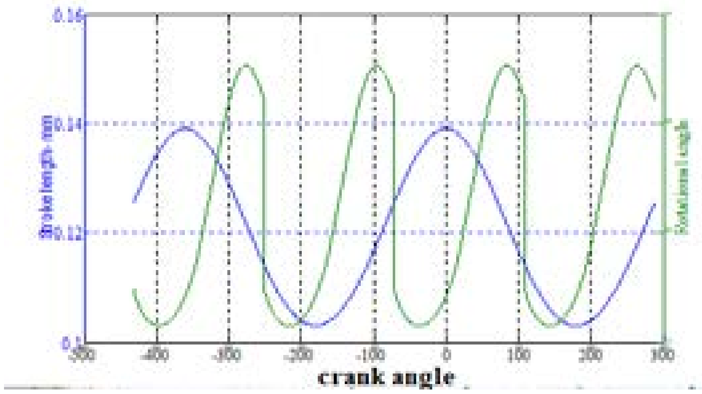

Figure 12 *Variations of the rotation motion of a skirt (Case 1)*

simulate the piston velocity for given testing conditions, as shown in Figures 6 and 7.

Using Equation (6), the mobility was computed, e as seen in Figures 8 and 9.

Similarly, the Mobility of a r cylinder block was computed using integration of accelerometer data as shown in Figures 10 and ,11.

Using the concept of anti-resonant frequency(ω_a), as discussed in previous section, various dynamic parameters of the liner-piston were computed for the given test conditions (Table 4), as seen in Table 5.

During the motion simulation, the bottom dead center positions (BDC) was taken as a reference point. The initial location of a piston is set at 0 mm as the bottom boundary of liner and the upper boundary of the cylinder liner is set at 0.5 mm, which is a clearance between the skirt and a liner.

As seen from Figure 12, the piston tilting angle changes its direction at both dead centers. In order to visualize the pistons secondary motion during the reciprocating motion, the piston secondary motion is represented in a graphical form and the piston lateral motion and rotating motion are normalized to the piston stroke position, based on the reciprocating motion of a piston, as shown in Figure 13.

It is evident from the plot that the piston remains at the lower boundary cylinder liner for a longer time, as compared to the upper boundary of a cylinder wall. In addition, the piston is predicted to slide for a crank angle of 100 º before the TDC along the cylinder liner (Figure 13).

Figure 14 shows the measured vibratory response of the cylinder block in the vibration amplitudes, as captured by accelerometer. The vibration of the cylinder decays after the first impact of the piston on the upper boundary of a liner. The vibration is induced once

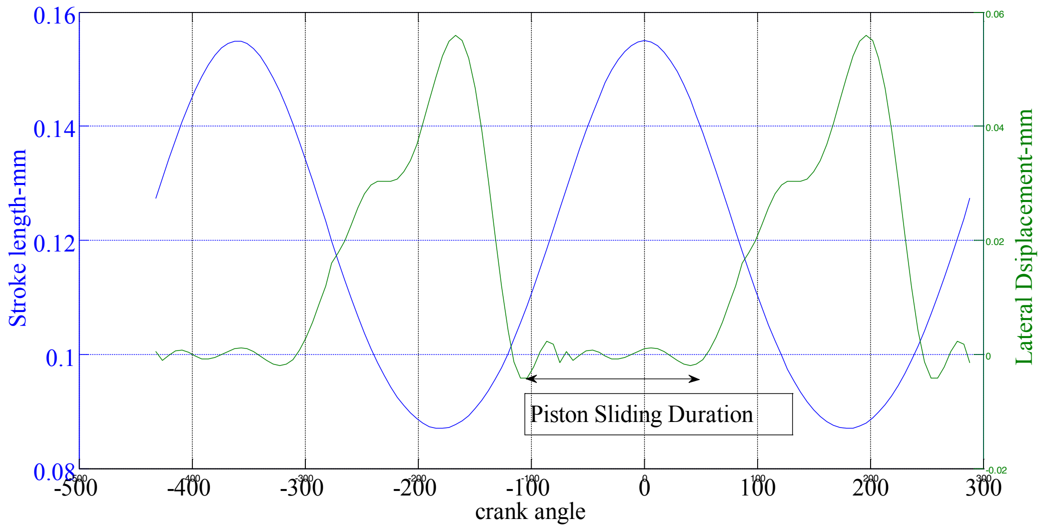

***Figure13** The piston's secondary motion (Case 1)*

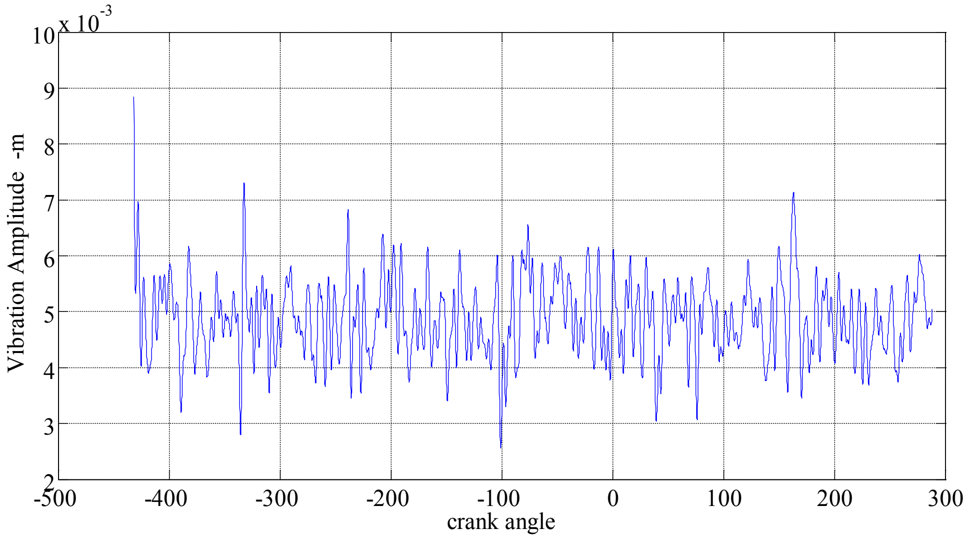

***Figure14** Vibration response of the engine block (Case 1)*

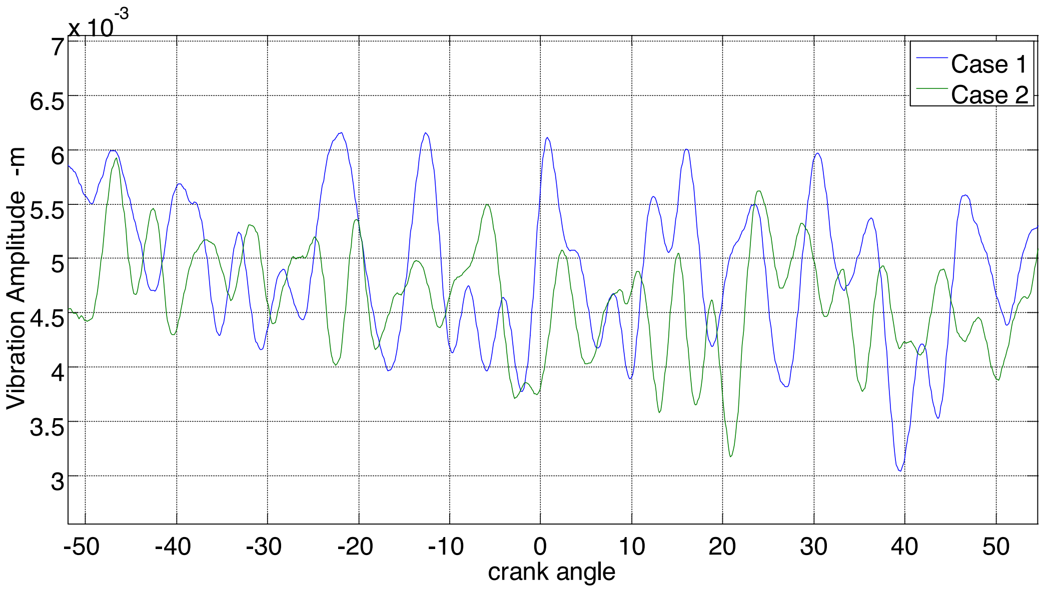

***Figure 15** Effects of variations of the engine speed on engine block vibrations (2000 rpm)*

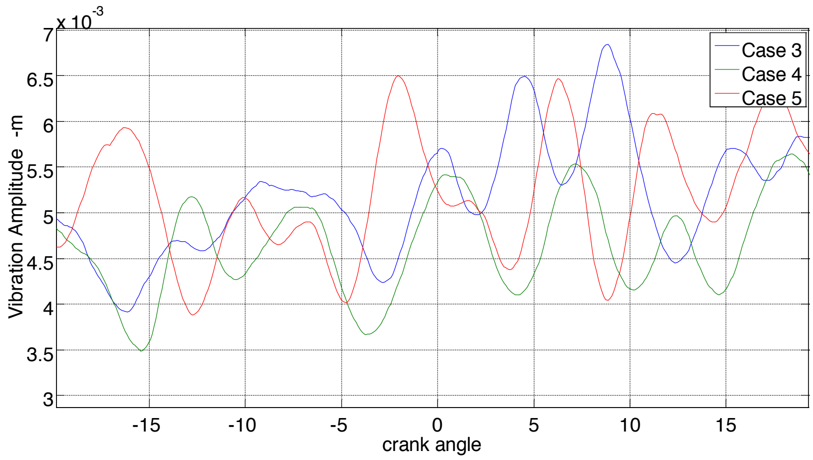

***Figure 16** Effects of variations of the engine speed on block vibrations (3000 rpm)*

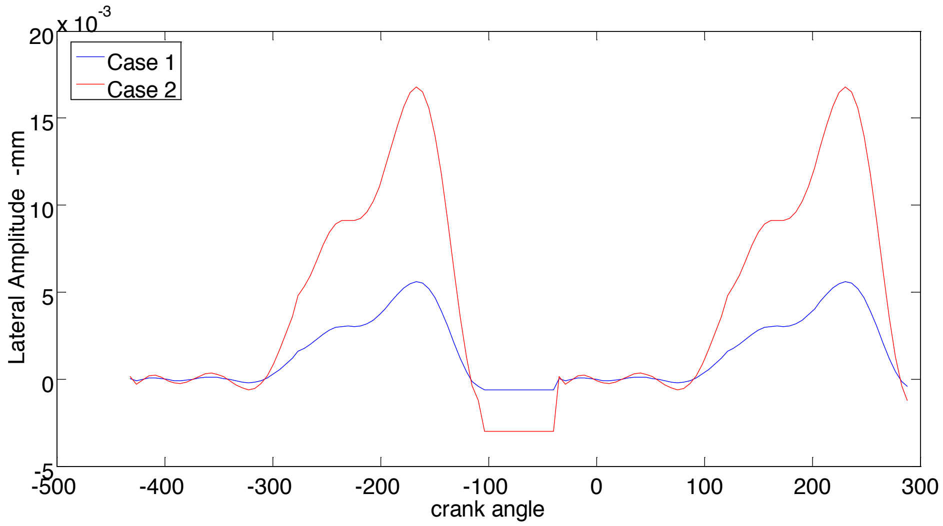

***Figure 17** Effects of variations of the engine speed on the secondary motion of a skirt (2000 rpm)*

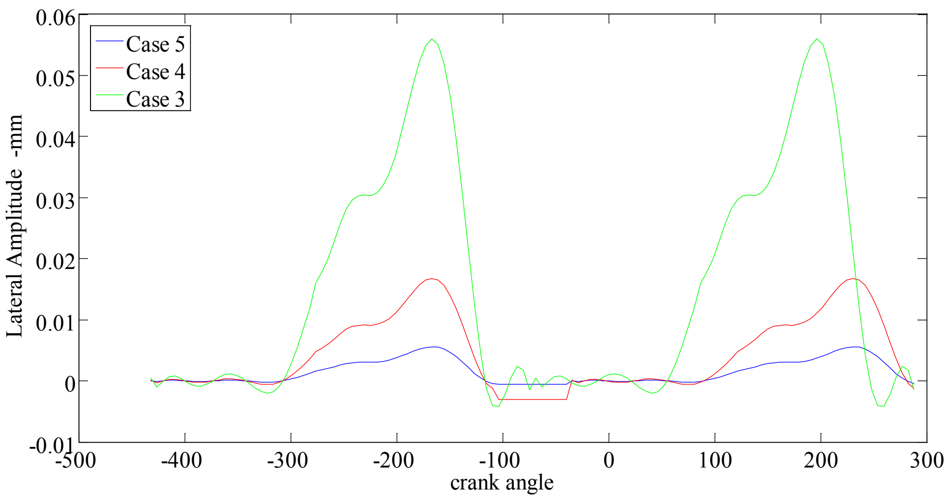

***Figure 18** Effects of variations of the engine speed on the secondary motion of a skirt (3000 rpm)*

again when the piston impacts lower cylinder liner. The induced vibrations had an amplitude of order of 7 $\times 10^{-3}$ m. As the engine operating speed increases, the piston side thrust force, which is a function of the engine rotating speed, increases. An increase in the side thrust force, acting on the piston, results in the piston bouncing off the cylinder liner more frequently at higher speeds, as seen from Figures 15 and 16.

The induced vibrations of a block also increase with engine speed. The sliding duration also falls with an increase in velocity as shown in Figures 17 and 18. This is due to the higher impact force and acceleration generated during the piston slap and the reaction impact force from a liner acting on the skirt increases. At the lower engine speeds, the vibration response of the cylinder block induced by the first slap of a piston has a longer duration to decay before the second slap occurs. However, with the speed increase, the vibration response of a block has the shorter duration of decay and response from the second slap is combined with the first one.

8 Conclusions

A lumped system model was discussed in the present paper. Various dynamic parameters of a system were calculated, using the concept of mobility, which were later used to simulate the lateral motion of a piston, as well as the resulting engine block vibrations. Values of the first resonance frequencies of both the skirt and a liner were found to be in 100 Hz-160 Hz range and it remains unaffected by variations in the engine operational conditions. Several peaks were found in the simulated block vibrations, which were related to impacts of a skirt with liner. The COMSOL-7 software was then used to analyze the tilting motion of a piston, which showed a good match with that simulated by solving dynamic equations of motion. Effects of load and speed on lateral motion of piston skirt were also investigated. The piston skirt was also found to slide along a liner a few crank angle degrees before the TDC position. This sliding motion was less dominant during the power stroke as the bouncing motion dominates the dynamic motion of a skirt. The duration of sliding motion of a piston along the liner was observed to decrease with increase in load and speed conditions, which is in agreement with previous available literature.

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