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.

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 Turnbull 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).

For condition of no impact, (X_p - X_b = X_c) the motion was governed by Equation (1).
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 (\(\mu\)), number of piston rings (\(n\)) and the shear area of a contact (\(A_s\)) as \([27-57]\):

\[
F_f = \mu V A_s 1/h, \quad (3)
\]

\[
F_{fr} = n\mu V A_s 2/h, \quad (4)
\]

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 \(V(J\omega)\) of a structure to exciting force \(F(J\omega)\) acting on a structure \([5]\):

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

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

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

\[
M(J\omega) = -J/M\omega^2, \quad (7)
\]
Above the anti resonance frequency, the point mobility can be written as:

\[ M(\omega) = -\omega J \omega / K. \]  

(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

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**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 | \( P_{\text{injection}} \) (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)
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.

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.

Table 5 Dynamic features of a system

| Test case | Piston parameter | Liner parameter |
|-----------|------------------|-----------------|
| 1         | \(\omega_a\) 100 Hz | \(\omega_a\) 39 Hz |
|           | \(C_p\) 109330 (kg/s) | \(C_p\) 42884 (kg/s) |
|           | \(K_p\) 174 (kg/s²) | \(K_p\) 175 (kg/s²) |
|           | \(m_p\) 174 (kg) | \(m_p\) 175 (kg) |
| 2         | \(\omega_a\) 100 Hz | \(\omega_a\) 39 Hz |
|           | \(C_p\) 109330 (kg/s) | \(C_p\) 109330 (kg/s) |
|           | \(K_p\) 174 (kg/s²) | \(K_p\) 174 (kg/s²) |
|           | \(m_p\) 174 (kg) | \(m_p\) 174 (kg) |
| 3         | \(\omega_a\) 158 Hz | \(\omega_a\) 63 Hz |
|           | \(C_p\) 172750 (kg/s) | \(C_p\) 69669 (kg/s) |
|           | \(K_p\) 174 (kg/s²) | \(K_p\) 174 (kg/s²) |
|           | \(m_p\) 174 (kg) | \(m_p\) 176 (kg) |
| 4         | \(\omega_a\) 158 Hz | \(\omega_a\) 63 Hz |
|           | \(C_p\) 109330 (kg/s) | \(C_p\) 109330 (kg/s) |
|           | \(K_p\) 174 (kg/s²) | \(K_p\) 174 (kg/s²) |
|           | \(m_p\) 174 (kg) | \(m_p\) 174 (kg) |
| 5         | \(\omega_a\) 158 Hz | \(\omega_a\) 63 Hz |
|           | \(C_p\) 172750 (kg/s) | \(C_p\) 69669 (kg/s) |
|           | \(K_p\) 174 (kg/s²) | \(K_p\) 174 (kg/s²) |
|           | \(m_p\) 174 (kg) | \(m_p\) 176 (kg) |

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° 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
Figure 13 The piston’s secondary motion (Case 1)

Figure 14 Vibration response of the engine block (Case 1)

Figure 15 Effects of variations of the engine speed on engine block vibrations (2000 rpm)
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
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
Values of the first resonance frequencies of both the skirt and a liner were found to be in the 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.

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.
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