Ant Colony Optimization for the Minimization of Internal Combustion Engine Forces and Displacements

T. Ramachandran, K. P. Padmanaban and J. Vinayagamoorthy

Abstract The unbalanced forces from the engine reciprocating and rotating component are the major contributors for the internal combustion engine vibration. The reciprocating and rotational inertial forces are modeled to determine the total forces excited and displacements caused at the engine mounts. A heuristic optimization (ACA) is used to optimize the design variables to reduce the forces and displacements at the supports.

Keywords IC engine vibration • Inertial force • ACA

1 Introduction

The reasons for the internal combustion (IC) vibration are fluctuating forces at the skirt of the piston and at the reciprocating components [1]. Impact forces at piston, due to the unbalanced forces of the rotating and reciprocating parts, are the common impact phenomenon existing in the reciprocating IC engine. It also plays an important role in the transverse vibration of the engine [2]. In the crank and slider mechanism of the IC engine, the piston is driven by the cylinder gas pressure and moves inside the cylinder along with the connecting rod and the crankshaft [3]. At the same time the periodical changing of direction of the connecting rod leads to the variation of inertial force in the cylinder wall and crank shaft. The unbalanced forces from the piston and connecting rod setup create not only the variational

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impact force on the cylinder walls but also the variational speed fluctuation on the crank shaft. So, the cylinder block experiences the vibrational force due to the impact force by slap as well as the speed variation in the crank shaft.

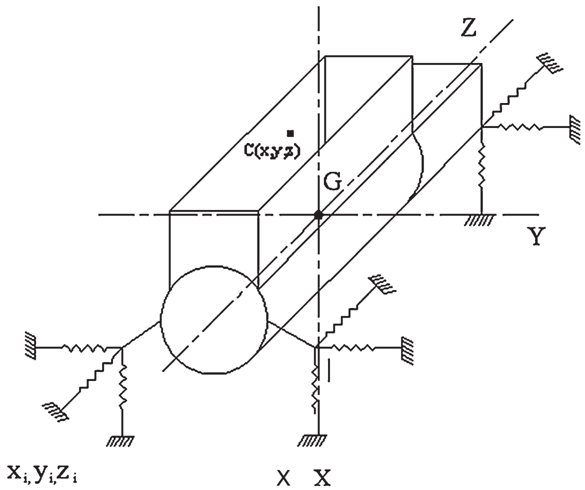
The solutions attempted to reduce the engine forces and displacements due to the diesel excitations are usually to isolate the engine from the vibratory forces. In this chapter, a 4-cylinder IC engine is considered as rigid body and the mathematical modeling is done to determine the forces and displacements exerted by the engine during the operation. Attempts are made to reduce the engine forces and displacements by optimizing the crank angles of the counter weights. Ant colony optimization [4] technique is used to predict the optimum positions of counter weights under minimum displacements at the engine supports.

2 Rigid Body Modeling of the Engine

The engine vibration can be classified into two types: vibration of engine components with respect to each other, named as internal vibration, and the vibration of engine block as a total, named as external vibration. The unbalanced forces from the engine components lead to the external vibration [1]. To analyse the engine vibration the engine will be considered as rigid body with 6° of freedom in *x*, *y* and *z* axis about its centre of gravity [5].

Consider an engine as a rigid body of mass M connected to a rigid chassis by three rubber engine mounts, as shown in Fig. 1. The origin of the fixed global coordinate system is located at the center of mass of engine. The *z*-axis is parallel to the crank shaft, and the *x*-axis is in the vertical direction. The general translation [6] is in the *y*-direction.

Fig. 1 Rigid body model of the engine



F_x, F_y and F_z are the sum of forces acting on the engine block in the x, y, and z direction, respectively and x, y, and z denote position of center of gravity of engine block. The engine's rotational equations of motion become principal axes of inertia; therefore, the products of inertia are eliminated. Three kinds of forces and moments act on the engine block: (1) shaking forces (inertia forces) and moments generated by moving links of the slider-crank mechanism, (2) forces and moments associated with the balancing system, and (3) reaction forces exerted to the engine block by three engine mounts.

2.1 Forces and Displacements on Engine

The engine is supported by three mounts positioned between the engine and the supporting structures. To achieve decoupling of the roll mode, the engine mounting system is designed to allow the elastic axis of the isolators to coincide with the engine roll inertia axis. This technique is achieved by inclining the mounts. In order to achieve the inclination of the engine mounts to be effective, the compression rate of the mounts must be significantly higher than the shear rate. The relation between the inclination angle of an isolator and the angle of elastic axis is kept equal. The mounts considered in this model are standard viscoelastic material with the stiffness of the mounts [7] are assumed to constant in the three axis and for the entire analysis. The components of the engine assumed as rigid bodies and the force and displacements of the components during the cyclic motion of the engine was calculated by the kinematic and dynamic analysis of the crank-slider arrangement.

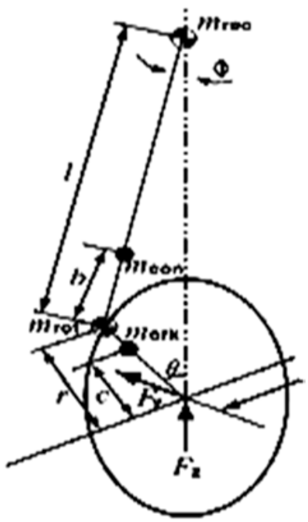
In this work a three dimensional model is proposed for the 4-cylinder diesel engine as shown in Fig. 1. This shows the piston motion in the directions along x, y, and z axis. The forces developed by the engine components in the cylinder are transferred to the engine supports as three dimensional forces and displacements. The translation forces from the cylinder are given by

$$\vec{F} = \begin{bmatrix} \sum F_x \\ \sum F_y \\ \sum F_z \end{bmatrix} \tag{1}$$

From the piston and connecting rod arrangement as shown in Fig. 2, the mass of the crank and the mass of the connecting rod are assumed to be concentrated at the mass center of the connecting rod. The masses of rotating and reciprocating components are calculated [8] as

$$m_{\mathrm{rot}} = \left({}^{c}\!/_{l}\right) m_{\mathrm{crk}} + \left(\frac{l-b}{l}\right) m_{\mathrm{con}} \tag{2}$$

Fig. 2 Crank and slider arrangement



$$m_{\text{rec}} = \left({}^{b}/_{l}\right) m_{\text{con}} + m_p \tag{3}$$

The forces from the engine cylinder [9] are calculated by adding the gas force and piston force from the reciprocating components and resolved into the vertical and horizontal [10] (radial and tangential) components.

x-directional force exerted at the ith cylinder

$$\overrightarrow{F}_{\text{x,i}} = (F_{\text{rot}} + F_{\text{pr}} + F_{\text{rd}})_x \overrightarrow{e_x} + (F_{\text{rt}} + F_{\text{rec}})_{\text{y}} \overrightarrow{e_y}; \tag{4}$$

y-directional force exerted at the ith cylinder

$$\overrightarrow{F_{y,i}} = (F_{\text{rot}} + F_{\text{rec}})_y \overrightarrow{e_y} \tag{5}$$

Gas pressure exerted at the ith cylinder

$$F_{\text{pr,i}} = {}^{\pi}/_{4} d^2 (a_1 \cos k\theta + a_2 \sin k\theta) \overrightarrow{e_y} \tag{6}$$

Unbalanced force exerted at the counter weight disc in the x direction

$$F_{\text{rd},x} = m_{\text{rd}} r_{\text{rd}} \omega^2 \sin(\theta + \psi); \tag{7}$$

y-directional force due to rotary components

$$\overrightarrow{F}_{\text{rot},y} = m_{\text{rot}} r \omega^2 \cos\theta; \tag{8}$$

Unbalanced force exerted at the counter weight disc in the y direction

$$F_{\mathrm{rd},y} = m_{\mathrm{rd}r_{\mathrm{rd}}}\omega^2 \cos(\theta + \psi); \tag{9}$$

y-directional force due to reciprocating components Table 1

$$F_{\mathrm{rec}} = m_{\mathrm{rec}} r\omega^2 \left(\cos\theta + 4A_2 \cos 2\theta + 16A_4 \cos 4\theta\right) \tag{10}$$

where,

$$A_2 = \frac{l}{r}\left[\frac{1}{4}({}^{l}/_{r})^2 + \frac{1}{16}({}^{l}/_{r})^4\right];$$
$$A_4 = -\frac{l}{r}\left[\frac{1}{64}({}^{l}/_{r})^4 + \frac{1}{256}({}^{l}/_{r})^6\right];$$

In finalising the model, the model allowances must be included which are must for the determination of dissipation of energy due to damping forces at the supports. Substituting the equation for the inertial forces and reaction forces [11] unbalance forces associated with the engine in terms of Newton law motion gives the following translational and rotational equations of motion of the system under the idling condition. The set of differential equations are solved numerically [12] over any respective time interval [0, T] to determine the displacements at the supports.

$$M\ddot{x} = \sum F_x$$
$$M\ddot{y} = \sum F_y$$
$$M\ddot{z} = \sum F_z$$

3 Ant Colony Algorithm for Optimization

Colonies of social insects can exhibit an amazing variety of complex behaviors and have always captured the interest of biologists and entomologists. The study of ant colonies behavior turned out to be very fruitful, giving rise to a completely novel field of research, now known as ant algorithms. In ant algorithms, a colony

Table 1 Force components

$F_{x,i}$, $F_{y,,i}$,$F_{z,i}$	x, y and z axis forces of the ith cylinder
F_{rot}	Forces from the rotational parts
F_{pr}	Forces due to the gas pressure
F_{rd}	Reaction forces at the rotating disc
F_{rec}	Forces from the reciprocating parts
θ	Crank angle
ψ	Connecting rod angle at the piston
a_1, a_2	Fourier constants

of relatively simple agents called as ants, efficiently carries out complex tasks such as resource optimization and control [13]. Ants deposit pheromones while carrying out their own tasks. These modifications change the way sensed by the other ants in the colony and implicitly act as a signal triggering other ants' behaviors that again generate new modifications that will simulate other ants and so on.

3.1 ACA-Based Engine Vibration Optimization Method

To apply the ant colony algorithm for the engine vibration optimization problems, randomly 'R' solutions are selected from different possible solutions. A critical value is fixed about which number of superior and inferior solutions are defined. Global search is carried for inferior solutions, whereas the local search is carried out for the superior solutions. In this chapter, the ant colony [4] algorithm is used in continuous engine vibration displacement optimization. The various processes involved in ant colony algorithm-based vibration displacement optimization method are as follows: (1) initialization, (2) global search, and (3) local search. Figure 2 explains the distribution of ants for local search and global search.

3.2 Initialization

1. 20 vibration displacements are randomly generated from the range defined for each vibration displacement model.
2. Then, the vibration displacement models are sorted according to ascending order of the solutions.
3. The solutions from 1 to 12 are named as superior solutions and from 13 to 20 are named as inferior solutions.

3.3 Global Search

The global search is done to improve the inferior solutions. This search includes crossover or random walk, mutation and trail diffusion.

Crossover or random walk In this process, the inferior solutions from 13 to 18 are replaced by the superior solutions. This process includes the following steps:

(a) Replacement of each inferior solution by a superior solution is decided based on the crossover probability.
(b) To replace 13th solution, a random number between 1 and 12 is generated. Then, the corresponding solution in the superior region replaces the 13th inferior solution.

(c) The selected solutions in the superior region should be excluded, so that it is not selected again for replacement.
(d) The above procedure is repeated up to the 18th solution.

Mutation The mutation process further improves the replaced solutions. This process includes the following
steps:

(a) The crank angle of each model in the replaced 13th model modified by adding or subtracting with mutation step size (Δ).

The mutation step size (Δ) is obtained as

$$\Delta = R(1 - r^{(1-T)^b}) \tag{11}$$

where,

$$R = (X_j\text{max} - X_i)$$

$X_{\max}$ maximum range of crank angle defined for the cylinders
X_i angle of the respective cylinder
R random number
T ratio of current iteration to the total no of iteration
B constant (obtained by trial)

(b) The mutation probability (Pm) is set. Then a random number is generated between 0 and 1. If the random number generated is less than Pm, the mutation step size (Δ) is subtracted to the node number of the respective displacement configuration or else it is added to the node number of the respective crank angle configuration element. The same procedure is repeated up to the 18th solution.

Trail diffusion The trail diffusion improves the 19 and 20th solutions. This process includes the following steps.

(a) Two displacements are randomly selected from the superior solutions, and they are named as parent-1 and parent-2. The crank angle of each configuration in parent-1 and parent-2 is termed as XP1 and XP2, respectively. The new set angles obtained from parent-1 and parent-2 is termed as Child. The angle of each configuration in the Child configuration is termed as XC.
(b) One more random number (α) is generated between 0 and 1 for the angle of each crank angle configuration.
(c) If α is between 0 and 0.5, then the new crank angle position of each configuration element of the new layout is obtained by $\mathrm{XC} = (\alpha)\mathrm{XP}_1 + (1-\alpha)\mathrm{XP}_2$
(d) If α is between 0.5 and 0.75, then the new position of each cylinder of the configuration is obtained by $\mathrm{XC} = \mathrm{XP}_1$

(e) If α is between 0.75 and 1, then the new angle position of each cylinder of the new configuration is obtained by XC $= XP_2$
(f) The above procedure is repeated for the 20th solution also. After the crossover or random walk, mutation, and trail diffusion processes, the solutions for the modified configurations from 13 to 20th are found using calculations.

3.4 Local Search

The local search is done to improve the superior solutions from 1 to 12.

This process includes the following steps.

The average pheromone value is calculated by $P_{\text{avg}} = \sum P / N_s$

where,

P pheromone value of each solution (Initial value is assumed to be 1.0)

N_S number of superior solutions

(a) A random number is generated between 0 and 1. If the number generated is less than average pheromone value (P_{avg}), the search is further pursued or else the ant quits, and then leaves the solution without any alteration.
(b) A limiting step value LS, which is added to the number of the respective cylinder when the random number generated is greater than 0.5 and subtracted to the number of the respective cylinder when the random number generated is less than 0.5, is calculated as follows: $L_S = K_1 - \mathrm{A} \times \mathrm{K}_2)$ where, K_1 and K_2 are the values chosen such that $K1 > K2$. 'A' is the age, which is assumed to be 10 for all the solutions in the first iteration.
(c) All the crank angles corresponding to the superior solutions are modified by local search and solutions for the modified angles from 1 to 12 are found using model calculations.
(d) The new age for each solution for the next generation is calculated as follows:

If the current solution is less than the previous solution, the age for the new solution is calculated as follows:

$$A = A_{i-1} + 1$$

If the new solution is greater than the previous solution, the age for the new solution is calculated as follows:

$$A = A_{i-1} - 1$$

where,

A_i is the age for the new iteration, A_{i-1} is the age for the previous iteration.

(e) The new pheromone value of the ant in the next iteration is also calculated as follows:$P_i = \frac{S_i - S_{i-1}}{S_{i-1}} + P_{i-1}$

where,

P_i—pheromone value for the new solution, S_i—the value of current solution,
P_{i-1}—pheromone value for the old solution, S_{i-1}—the value of old solution.

The above steps, i.e., local and global searches are performed in all the iterations to improve the solutions.

4 Convergence of ACA

For each crank angle considered in the particular ACA iteration, the forces are applied sequentially for the crank and slider arrangement of all cylinders. The maximum deformation (Δ_{max}) among the maximums for force application is found out. The same procedure is repeated for all the crank position for all the iteration. Then using ACA, the minimum deformation (Δ_{min}) among maximums for each crank angle is found. The same procedure is repeated for all the iterations (N_G). The algorithms converge if either the number of iterations over which no change in the objective function value is obtained, $N_{chg,}$ or if the number of iterations, N, reaches the defined maximum number of iterations, N_G whichever is earlier. The optimization of the crank position is carried out for the whole process in a single step. The different runs are performed in ACA-based continuous optimization method until the minimum engine displacement corresponding to optimal crank angle is determined.

5 Numerical Simulations and Results

The calculations of the mathematical model are performed using Matlab to determine the forces and displacements of the engine. To determine the engine forces the simulation of motion of engine over a given time interval for specific speed is made. The calculated time history of the behavior of the system enables the determinations of the objective function of the engine model (Table 2). In the simulation the initialization of crank angle of the any one cylinder is set to zero. If the first cylinder angle is set to zero, then the subsequent angle are assigned as design variables. The engine is simulated at a speed of 1,500 rpm for specified time duration of 0.05 s to obtain the forces and displacements exerted by the engine at the supports. In the AC optimization process the objective function is defined with the convergence parameters and then the optimization of the objective function is made after the predefined iterations. The results of every set of generations of each run are determined and the optimum values of the each design variables (Tables 3 and 4) with respect to the minimum forces and displacements (Figs. 3 and 4) are revealed. The minimum of minimum force and displacement of 5 runs is considered to be the optimum conditions of the engine variables. The force and displacement results of the model are compared with and without optimization. The results obtained in the ACA shows good convergence (Fig. 5).

6 Conclusion

In this research work, the forces and displacements caused in a 4-cylinder diesel engine are determined by considering the engine as the rigid model. To determine the forces and displacements at the engine supports the forces from the various engine components (rotating and reciprocating components) are modeled through the Newton law of motion by considering the different constants of the engine specifications. The force models of the all components are coupled together and made into the single force system by considering the force system as one way coupling model. By summing up the force components of the all cylinders into single system the final model is developed (Tables 2, 3, 4 and 5, and Figs. 3, 4 and 5).

The model developed is simulated with a help of Matlab such that the design variables are specified with a range and the response is the displacements at the engine mounts. Finally, the optimization of the design variables is made using the

Table 2 Engine parameters

Abbreviation	Parameter	Value
M	Mass of the engine	250 kg
R	Radius of the crank	0.02 m
L	Length of the connecting rod	0.11 m
m_{con}	Mass of the connecting rod	1.2 kg
m_{crk}	Mass of the crank shaft	2.5 kg
m_p	Mass of the piston	2.65 kg
N	Engine rotational speed	1500 rpm
K_x	Elastomeric mount stiffness in x axis	3.5E5 N/m
K_y	Elastomeric mount Stiffness in y axis	3.9E5 N/m
K_z	Elastomeric mount Stiffness in z axis	2.3E5 N/m

Table 3 Optimised crank angles for the counter weights

	Optimum crank angle	
Cylinder	Before using ACA	After using ACA
1.	0°	4°
2.	**255°**	**257.5°**
3.	105.3°	100.1°
4.	150°	156°

Table 4 Optimised transmitted forces

	Force transmitted (*N*)	
Force components	Before using ACA	After using ACA
F_x	421.5	400.22
F_y	2500.8	1800.6
F_z	130.13	90.64

Table 5 Optimised transmitted forces

Force components	Displacement (mm) Before using ACA	After using ACA
X	0.00534	0.0031
Y	−0.0296	−0.0283
Z	0.001051	0.00104

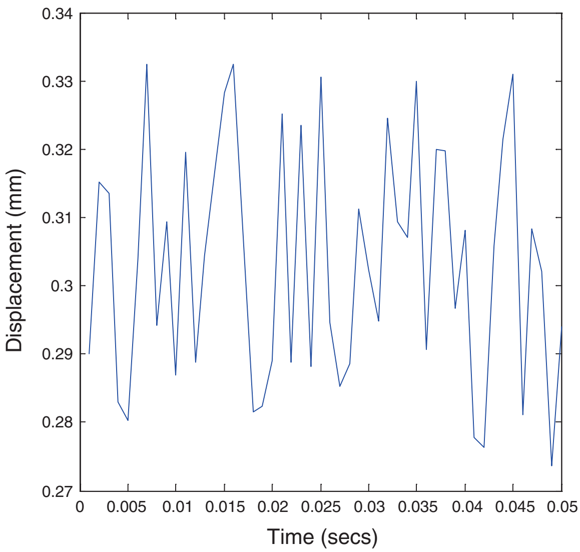


Fig. 3 Engine displacement variation without optimized values

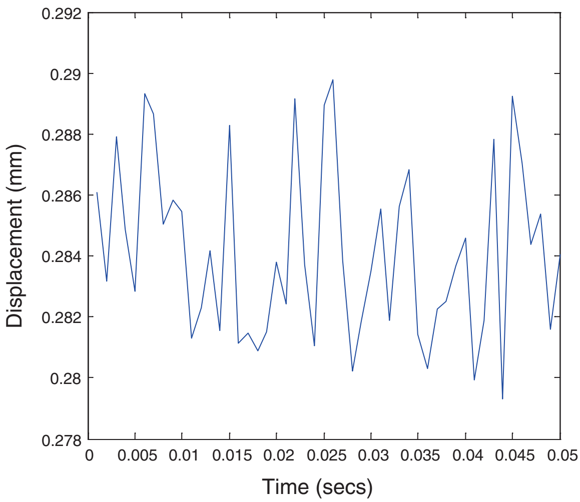


Fig. 4 Engine Displacement variation with optimised value the minimum displacement

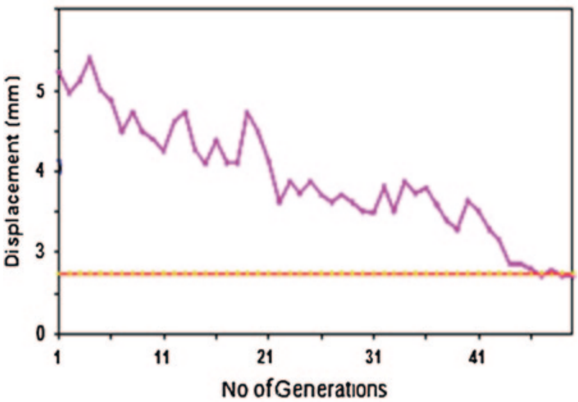


Fig. 5 Convergence of ACA towards

Ant Colony Optimization method to determine the minimum displacements and corresponding forces at the engine supports. In the AC optimization the convergence is obtained after 40 generations. The optimized parameters are once again simulated for the time duration of 0.05 s at a speed of 1500 rpm and the results of the optimized and non-optimized values of displacements are compared. The comparison shows the improvement in the reduction of the displacements and forces at the mounts when AC optimizations is used.

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