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# Optimal characteristics determination of engine mounting system using TRA mode decoupling with emphasis on frequency responses

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ABSTRACT

It is possible to improve vehicle vibration by tuning the parameters of engine mounting system. By optimization of mount characteristics or finding the optimal position of mounts, vibration of the engine and transmitted force from the engine to the chassis can be reduced. This paper examines the optimization of 6-degree-of-freedom engine mounting system based on torque roll axis (TRA) mode decoupling, so that TRA direction coincides with one of the natural modes of vibration. This is achieved by determination of optimal location and stiffness of mounts. In order to find feasible results, physical constraints are taken into account in optimization process. A detailed procedure of optimization problem is explained. Finally, by comparing the frequency and time responses of the optimal design with the original configuration, it is concluded that TRA decoupling is a proper objective function in engine mounting optimization and can greatly improve the vibration behavior of the engine. Achieving decoupled system, the optimal configuration has a better chance of placing dominant natural frequency below the operation range. Also, the forces transmitted through the mounts are reduced noticeably in the optimal design.

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## 1. Introduction

Tuning of engine mount characteristics plays an important role in reducing automotive engine vibration [1-3]. Combustion forces that produce periodic excitation in engine are major sources of vibration. Engine mounts are used to support engine and isolate the chassis from these

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vibrations [4, 5].Engine mount parameters such as stiffness and location of each mount have great effects in engine vibration isolation. Therefore, it is of considerable interest to optimize engine mount characteristics.

In recent decades many researches have dealt with the various methods of engine mount optimization. Optimization through decoupling engine mounting system is a good starting point for proper vehicle design[6]. Jeong et al. [3] have investigated engine torque roll axis (TRA) decoupling method. They provide analytical solution for complete decoupling of TRA mode and demonstrate that only TRA mode decoupling is sufficient to make system fully decoupled. The authors have proposed the necessary condition of decoupling and derived the equations which satisfy the decoupling condition. Akanda and Adulla[7] studied a six-cylinder four-wheel drive vehicle using full-vehicle finite element model. TRA decoupling method was used to decouple the engine rigid body modes. Instead of analytical solution, they defined decoupling optimization problem and gradient based method was implemented to solve that. El Hafidi et al.[8] compared three different decoupling techniques for engine mounting system to minimize vibration of powertrain in city buses. Kolte et al.[9] developed an optimization tool using Particle Swarm Optimization algorithm to optimize the automotive powertrain mounting system based on TRA decoupling and considered new constraints to the optimization problem. The optimization tool was performed for two different engines: a 625cc single-cylinder transverse mounted diesel engine of a load carrier and a 1.2L three-cylinder transverse mounted passenger vehicle diesel engine. In [10] the authors used decoupling method to improve engine abnormal low frequency vibration. Angrosch et al.[11] dealt with implementation of two decoupling concepts by applying numerical optimization of the modal kinetic energy distribution. Some researchers have analytically investigated the powertrain mode decoupling problem by considering the influence of varying mount properties including stiffness and damping[12, 13]. In addition, some researchers have been interested in coupled powertrain and frame system with consideration of dynamic interactions between the powertrain and other sub-systems [14, 15]. Sakhaei and Durali[16] presented a method for simulation of vehicle interior vibration. In this regard, a multilevel transfer path analysis using chain of transfer functions of the vehicle sub systems was employed to identify vibration transfer paths and rank them. Then, vibration behavior at vehicle interior target points was used as an index to optimize engine mount parameters. Based on the obtained results for vibration share of each mount, some modifications were proposed to decrease the vibration at target point. Since the mentioned modifications may have adverse effects on engine movements, dynamic analysis of the vehicle was performed using MSc. ADAMS software to evaluated engine movements in extreme drive maneuvers. Rasekhipour and Ohadi [17] presented optimization of hydraulic engine mounts (HEMs) based on a ride comfort index, i.e. vertical acceleration at driver position. In this regard, they considered and investigated three different models. The first model was a 13-degree-of-freedom (13-DOF) vehicle model including engine mounted to chassis via three HEMs. A 6-DOF engine mounted to the ground via three HEMs was considered as the second model and the third model included a simplified 1-DOF body mounted to the ground via one HEM. Then, the Directed Tabu Search optimization method was employed for optimization process. Finally, optimization results for the mentioned three models were presented and compared. Considering the correct constraints based on practical aspects, is one of the important steps in optimization of engine mounting system. However, some literatures have not taken appropriate constraints in optimization problem. In this paper, new constraints according to engine operational conditions have been proposed for optimizing the engine mounts' parameters to achieve feasible results. To the best of the authors' knowledge, constraints proposed in this study have not been presented in the literature in the past. On the other hand, employing accurate models have a considerable role in optimization problems. However, some studies have employed the simplified model of the engine for optimization process. For example, in some literatures, 1-DOF engine model has been considered or damping of the mounts has been neglected in optimization process. In this research, a relatively accurate 6-DOF engine model including damping parameters for the mounts is used for optimization purpose. The employed 6-DOF relative accurate engine model together with new constraints used in this study, result in a more perfect and exact optimization to achieve feasible results. This matter has not been presented in available literature. In addition, in this study the Genetic Algorithm (GA) is employed that has not been used in the literature for optimization of TRA decoupling.

In this study, TRA optimization problem is solved using GA optimization method because of its advantages in solving multi-dimensional optimization problems. GA is a useful optimization technique which is based on natural selection and evolution. The GA has some advantages that overcome limitations of other optimization techniques. It is a robust searching tool which does not break down in face of slight changes of inputs or noises and also can avoid local minima. It requires minimum problem information and has significant benefits in solving multi-dimensional problems. Moreover, it produces a list of solutions instead of single solution [18].

In this paper, optimization of engine mounting system based on TRA mode decoupling is investigated. By mode decoupling, the transmitted force from the engine to the chassis is reduced. In addition, mode decoupling guarantees that the input load excites only one pure mode with one natural frequency. Therefore, only one natural frequency is needed to be away from excitation frequency. This can be achieved by determination of optimal location and stiffness of mounts. By solving the optimization problem, acceptable mode decoupling along TRA is obtained. The optimal design is compared with the original configuration of mounting system. The obtained results in this paper indicate that TRA mode decoupling is a good criterion for optimization of engine mounting system and will result in improvement of engine vibration behavior. In section 2, equations of motion are presented. Concept of TRA mode decoupling is discussed in section 3. The optimization problem including objective function, optimization variables and constraints are presented in section 4. Finally, in section 5, optimization results, natural mode shapes and frequency and time responses of engine movements and transmitted forces to chassis are presented. According to the obtained results, the transmitted force decreases more than 50 percent for optimal system compared with original one. Conclusions are presented in section 6.

## 2. Equations of motion

The engine is modeled as a 6-DOF rigid body supported by mounts. An inertial coordinate system with the origin located in engine center-of-gravity (CG) is considered. It is assumed that x-axis is parallel to the crankshaft axis (Fig. 1). The 6-DOF model of engine on mounts in CG coordinate is presented as follows:

$$M\begin{bmatrix}\ddot{x}\\ \ddot{y}\\ \ddot{z}\\ \ddot{\theta}_x\\ \ddot{\theta}_y\\ \ddot{\theta}_z\end{bmatrix}+C\begin{bmatrix}\dot{x}\\ \dot{y}\\ \dot{z}\\ \dot{\theta}_x\\ \dot{\theta}_y\\ \dot{\theta}_z\end{bmatrix}+K\begin{bmatrix}x\\ y\\ z\\ \theta_x\\ \theta_y\\ \theta_z\end{bmatrix}=F \tag{1}$$

where $K$ and $C$ are stiffness and damping matrices, respectively, and $F$ is excitation force vector. Also, $M$ is mass matrix of engine given by:

$$M=\begin{bmatrix}m&0&0&0&0&0\\0&m&0&0&0&0\\0&0&m&0&0&0\\0&0&0&I_{11}&I_{12}&I_{13}\\0&0&0&I_{21}&I_{22}&I_{23}\\0&0&0&I_{31}&I_{32}&I_{33}\end{bmatrix} \tag{2}$$

where $m$ is engine mass and $I$ is mass moment of inertia about CG.

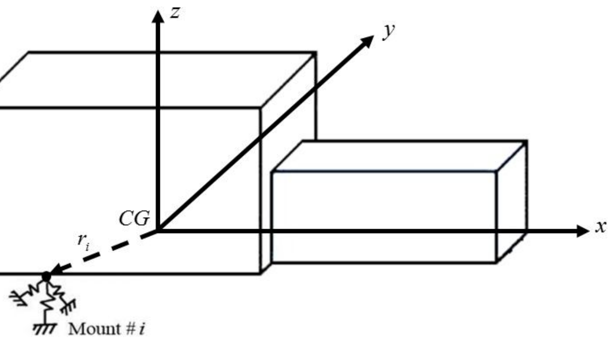



**Fig .1.** Definition of CG coordinate system

Each mount is modelled by three linear springs and dampers. It is considered that $K_{mi}$ and $C_{mi}$ are the stiffness and damping matrices of $i$th mount in its local coordinate system (the coordinate system located in $i$th mount and is parallel to inertial coordinate system), respectively, as follows:

$$K_{mi}=\begin{bmatrix}k_{mxi}&0&0\\0&k_{myi}&0\\0&0&k_{mzi}\end{bmatrix} \tag{3}$$

$$C_{mi} = \begin{bmatrix} c_{mxi} & 0 & 0 \\ 0 & c_{myi} & 0 \\ 0 & 0 & c_{mzi} \end{bmatrix} \tag{4}$$

In addition, $r_i = \begin{bmatrix} r_{xi} & r_{yi} & r_{zi} \end{bmatrix}^T$ is the position vector of $i$th mount in CG coordinate system. The total stiffness matrix $(K)$ is obtained using local stiffness matrix $(K_{mi})$ and position vector of each mount $(r_i)$ as demonstrated in:

$$K = \sum_{i=1}^{n} \begin{bmatrix} K_{mi} & K_{mi} R_i^T \\ (K_{mi} R_i^T)^T & R_i K_{mi} R_i^T \end{bmatrix} \tag{5}$$

where $i$ is the mount index and $R_i$ is rotation matrix represented as follows[3]:

$$R_i = \begin{bmatrix} 0 & -r_{zi} & r_{yi} \\ r_{zi} & 0 & -r_{xi} \\ -r_{yi} & r_{xi} & 0 \end{bmatrix} \tag{6}$$

Also, total damping matrix $(C)$ is obtained in the same way.

Fakhari and Ohadi[19] have developed a comprehensive dynamic model of EF7 engine on the mounts. EF7 engine is a four-cylinder engine supported by three mounts as shown in Fig. 2. More detailed specifications of EF7 engine such as engine mass properties, stiffness and damping of the mounts can be found in reference[19]. In this study, a four-point engine mounting system is considered for EF7 engine and optimization problem of the mounting system is investigated. Engine and mounts' parameters are given in Table 1.

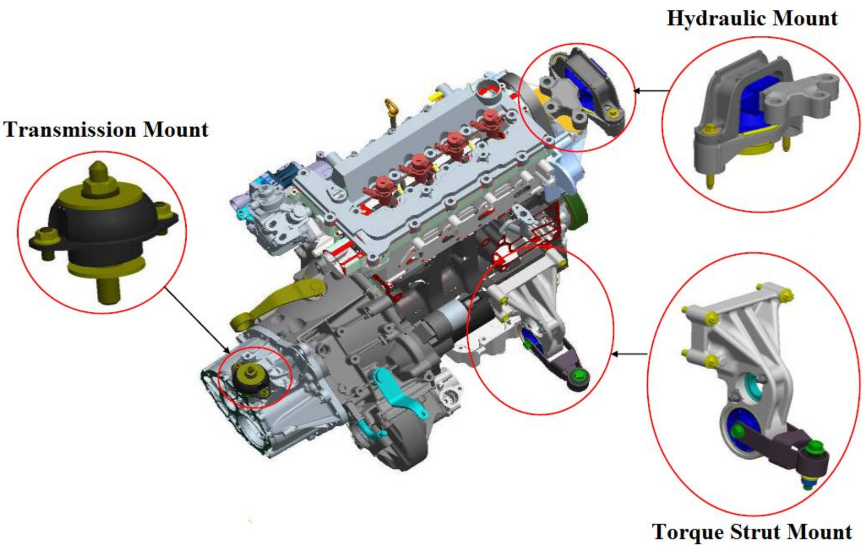



**Fig. 2**. EF7 Engine and its mounts

**Table 1.** Original mounting system parameters

| Engine parameters | | | | |
|---|---|---|---|---|
| | Mass ($kg$) | | Mass moment of inertia with respect to CG coordinate system ($kg$ m$^2$) | |
| | $m = 204.01$ | | $I = \begin{bmatrix} 6.84 & -1.05 & 1.96 \\ -1.05 & 12.5 & -0.261 \\ 1.96 & -0.261 & 10.5 \end{bmatrix}$ | |
| Mount parameters | Mount# | | | |
| | 1 | 2 | 3 | 4 |
| Stiffness (N/m) | | | | |
| $k_{mx}$ | 50,000 | 95,000 | 63,050 | 50,000 |
| $k_{my}$ | 150,000 | 120,000 | 277,690 | 150,000 |
| $k_{mz}$ | 350,000 | 280,000 | 222,220 | 350,000 |
| Location with respect to CG coordinate system (m) | | | | |
| $r_x$ | -0.4047 | 0.4523 | -0.1327 | -0.1327 |
| $r_y$ | -0.0201 | 0.0592 | 0.2162 | -0.2136 |
| $r_z$ | 0.1870 | 0.0502 | -0.2519 | -0.2519 |

## 3. TRA decoupling concept

TRA is an axis around which unconstrained engine will rotate when a torque is applied about an arbitrary axis. By neglecting all second order terms in Euler equations of motion, TRA will have a unique direction that depends only on inertial properties of the engine and the direction of applied torque. The main excitation of the engine is about crankshaft axis which produces the main rotation of the engine. Then, as the unconstrained engine (without mounts) is excited by the torque applied along crankshaft axis (was considered $x$-$axis$ in this study), the rotation of the engine will be about TRA[3].

It is assumed that $\theta_G = [\theta_x \quad \theta_y \quad \theta_z]^T$ represents rotation of the engine in CG coordinate system and the harmonic torque $T_x(t) = [T_x e^{i\omega t} \quad 0 \quad 0]^T$ is applied to the engine. The simplified Euler equations of motion are given by:

$$[I]\ddot{\theta}_G(t) = [T_x e^{i\omega t} \quad 0 \quad 0]^T \tag{7}$$

Hence the response $\theta_G(\mathrm{t})$ will be as below[3]:

$$\theta_G(t) = [I]^{-1}[T_x \quad 0 \quad 0]^T \frac{1}{\omega^2} e^{i\omega t} \tag{8}$$

As the response of $\theta_G$ is about TRA, the unique TRA direction would be as follows (based on equation (8)):

$$\bar{\theta}_{TRA} = a[I]^{-1}[1 \quad 0 \quad 0]^T \tag{9}$$

where $a$ is normalizing factor.

The TRA direction in generalized coordinate system is defined as $q_{TRA} = \begin{bmatrix}0 & 0 & 0 & \bar{\theta}_{TRA}^T\end{bmatrix}^T$. In order to deal with TRA decoupling problem, it is better to define a new coordinate system $R_{TRA}$ (with $\acute{x}$ , $\acute{y}$ and $\acute{z}$ axes) which is located in CG and $\acute{x}$- axis is parallel to TRA direction. One can define TRA direction in $R_{TRA}$ coordinate system as $q'_{TRA} = [0 \quad 0 \quad 0 \quad 1 \quad 0 \quad 0]^T$ .

The corresponding eigenvalue problem for proportionally damped mounting system is given by [20]:

$$M^{-1}Kq_{TRA} = \lambda q_{TRA} \tag{10}$$

where $\lambda$ is eigenvalue.

The complete decoupling of TRA mode is achieved when Eq. (10), which leads to six equations, is satisfied. In other words, the TRA must be one of the natural modes of vibration. The complete decoupling of modes is not possible due to the strong constraints. So, to achieve the decoupled system, the TRA decoupling procedure would be cast as a multi-objective optimization problem instead of solving the mentioned analytical equations.

## 4. The optimization problem

For the dynamic system under excitation force, the system response is sum of the natural modes which depends on excitation frequency. While for the decoupled system that is excited by one mode, the response would be along the mentioned mode independent of the frequency.

If TRA mode coincides with one of the natural modes of vibration, the system response would be a pure rotation about TRA. Since TRA direction relies only on the inertial properties of the engine and excitation direction (which is along crankshaft axis), system response has a fixed direction for decoupled system irrespective of the excitation frequency[3] .Therefore, in this study, the excitation (applied torque about crankshaft axis) is considered with a constant frequency and unit amplitude.

### *4.1.Objective*

The purpose of optimization is to force one of the natural modes to coincide with TRA direction. It is achieved if the engine response becomes a pure rotation about TRA by torque excitation along the crankshaft. Therefore, the criterion of TRA decoupling is to make the engine response be parallel to TRA. So, the optimization objective is considered to minimize the error between the rotation axis of the engine and TRA direction in the presence of torque excitation along crankshaft[8].

Thus, the optimization target is formulated as following:

$$\left\{\begin{array}{c}\text{Target 1}: x=0; \quad \text{Target 2}: y=0; \quad \text{Target 3}: z=0; \quad \text{Target 4}: \bar{\theta}_G.\bar{\theta}_{TRA}-1=0 \\ \textit{in the presence of torque excitation along crankshaft}\end{array}\right\} \tag{11}$$

where $\bar{\theta}_G$ and $\bar{\theta}_{TRA}$ are normalized CG rotation vector and normalized TRA vector, respectively. So, the multi-objective optimization problem is considered to minimize four target equations as four objective functions (Obj1,..., Obj4). The mentioned objective functions are supposed as a single objective function (Eq. (12)) to make the optimization problem simple.

$$Obj = \frac{1}{4}\left( \left|\frac{Obj\ 1}{Obj\ 10}\right| + \left|\frac{Obj\ 2}{Obj\ 20}\right| + \left|\frac{Obj\ 3}{Obj\ 30}\right| + \left|\frac{Obj\ 4}{Obj\ 40}\right| \right) \tag{12}$$

where Obj10,…,Obj40 are the nominal values of Obj1,..., Obj4, respectively, obtained by original engine parameters. In other words, each objective function in above equation is normalized by its nominal values and is unity for the original system.

### *4.2. Optimization variables*

Stiffness and position of each mount are used as optimization variables. Sensitivity analysis is done to show the sensitivity of objective function to variation of each optimization variable. Then, variables with low sensitivity are eliminated to decrease the complexity of solving optimization problem. In addition, each stiffness variable is allowed to vary 50 percent of its original value in the optimization.

### *4.3. Constraints*

In order to find feasible optimization results, it is very important to consider limitations and constraints. Some constraints are applied in optimization process that are presented in the following subsections:

**Table 2.** Variation range of engine mounts positions according to physical constraints

| Position | Variation range (m) | Position | Variation range (m) |
|---|---|---|---|
| $r_{x1}$ | $[-0.1,0]$ | $r_{x3}$ | $[-0.1,+0.1]$ |
| $r_{y1}$ | 0 | $r_{y3}$ | 0 |
| $r_{z1}$ | $[0,+0.1]$ | $r_{z3}$ | $[-0.1,+0.1]$ |
| $r_{x2}$ | $[-0.1,+0.1]$ | $r_{x4}$ | $[-0.1,+0.1]$ |
| $r_{y2}$ | 0 | $r_{y4}$ | 0 |
| $r_{z2}$ | $[0,+0.1]$ | $r_{z4}$ | $[-0.1,+0.1]$ |

#### *4.3.1. Geometrical constraints*

*Due to limitations in the chassis for mount positioning, some physical constraints on position of the mounts should be considered. In this regards, the variation of some positions are considered to be 0.1* m*, while some position variables are considered fixed. Table 2 shows the possible variation range of position variables based on EF7 engine and related chassis.*

#### *4.3.2. Constraint on static deflections*

It is obvious that one of the main purposes of using engine mounts is to support the engine weight. If mounts have a low stiffness, they suffer from high static deflection which is caused by engine mass. So, Stiffnesses of mounts are chosen in a way that the static deflection of engine does not exceed the allowed limit[21].

Constraint1: Mounts static deflection <8mm

#### *4.3.3. Constraint on dynamic displacements*

It is ideal to use low stiffness mounts to reduce the transmitted vibration to the chassis. However, low stiffness mounts increase the vibration of engine and may cause collision between engine and chassis or damage delicate parts of the engine. Hence, it seems necessary to limit the dynamic displacement of the engine[21]. So, the maximum displacement of CG in $x$ , y and z directions is considered as a constraint in this optimization problem.

Constraint2: Dynamic displacement of Center of Gravity < 2 mm

#### *4.3.4. Constraint on the natural frequency*

In decoupled system, since the rotation about TRA is the dominant mode and the response of the system for all other modes is minimum, only one natural frequency becomes important. So, the natural frequency which belongs to the TRA mode must be placed below the engine excitation frequency range to prevent resonance phenomenon.

Constraint3 : TRA mode natural frequancy < 20 Hz

## 5. Results and discussion

In this study, optimization process using GA begins with 10 sets of candidate solutions and is repeated until 100 generations which is enough since it leads to solution convergence and covers a wide range of searching space. Using simple crossover genetic operation, a new population is formed. Since there are no precise criteria to define the GA's parameters, the optimization process has been irritated several times to fine the best answers.

According to the obtained results, the objective function decreases from initial value of 1 to 0.16 after performing the optimization, which indicates 84% reduction of the objective function. It is noted that complete decoupling of TRA mode is achieved when objective function has a value of 0.

By minimizing the objective function, the optimal values of parameters are obtained and presented in Table 3. The natural mode shapes of engine mounting system with original and optimal parameters are presented in Tables 4 and 5, respectively. Results clearly demonstrate that one of the mode shapes (4$^{th}$ mode) in optimal design has approached to TRA.

**Table 3.** Optimal parameters of engine mounts

| Stiffness (N/m) | | Location with respect to CG coordinate system (m) | |
|---|---|---|---|
| $k_{my1}$ | 212,500 | $r_{x1}$ | -0.4380 |
| $k_{mz1}$ | 495,830 | $r_{z1}$ | 0.2537 |
| $k_{mx2}$ | 47,500 | $r_{x2}$ | 0.5523 |
| $k_{my2}$ | 170,000 | $r_{x3}$ | -0.0327 |
| $k_{my3}$ | 138,840 | $r_{z3}$ | -0.2019 |
| $k_{mz3}$ | 222,220 | $r_{x4}$ | -0.0827 |
| $k_{mz4}$ | 291,670 | $r_{z4}$ | -0.2019 |

The frequency responses of CG movements in TRA coordinate system, for unit input torque along crankshaft, are presented for original and optimal mount characteristics in Fig. 3. The results confirm that optimal design of engine mounting system has a significant effect in frequency response of engine CG movements and it can be seen that acceptable decoupling has been occurred.

TRA mode decoupling leads to decrease in transmitted force to the chassis, since the rotation axis of constrained engine has approached to the rotation axis of unconstrained engine. Fig. 4 compares time response of total transmitted force to the chassis for original and optimal system. The result illustrates that the transmitted force decreases more than 50 percent for optimal system in comparison with original one. Also, Fig. 5 shows the frequency responses of the transmitted force through different mounts for applied torque along crankshaft. The obtained results show that transmitted forces to the chassis in the frequency range of engine operation are lowered for optimal system compared with original one.

**Table 4.** Natural mode shapes of original engine mounting system in TRA coordinate system

| Natural Mode | 1st 5.6 Hz | 2nd 7.9 Hz | 3rd 11.3 Hz | 4th 11.6 Hz | 5th 16.6 Hz | 6th 17.4 Hz |
|---|---|---|---|---|---|---|
| $\acute{x}$ | 1.0000 | -0.0307 | -0.0304 | 0.0672 | -0.0143 | 0.0179 |
| $\acute{y}$ | 0.1459 | -0.1783 | 0.0946 | -0.4676 | 0.0875 | -0.0248 |
| $\acute{z}$ | 0.1076 | 0.5761 | 0.0589 | -0.1186 | -0.0348 | -0.0693 |
| $\acute{\theta}_x$ | 0.0023 | 1.0000 | 0.5170 | 0.4038 | 1.0000 | 1.0000 |
| $\acute{\theta}_y$ | 0.2212 | -0.5951 | 1.0000 | 1.0000 | 0.0268 | -0.4957 |
| $\acute{\theta}_z$ | -0.0684 | -0.2195 | 0.4626 | -0.3683 | -0.9738 | 0.4086 |

**Table 5.** Natural mode shapes of optimal engine mounting system in TRA coordinate system

| Natural Mode | 1st 5.1 Hz | 2nd 9.1 Hz | 3rd 12.3 Hz | 4th 13.6 Hz | 5th 15.5 Hz | 6th 19.9 Hz |
|---|---|---|---|---|---|---|
| $\acute{x}$ | 1.0000 | -0.0803 | -0.1866 | **-0.0015** | -0.0026 | 0.0126 |
| $\acute{y}$ | 0.1479 | -0.3302 | 1.0000 | **0.0035** | -0.0108 | -0.0494 |
| $\acute{z}$ | 0.1298 | 1.0000 | 0.3158 | **0.0050** | 0.0105 | -0.0158 |
| $\acute{\theta}_x$ | -0.0067 | -0.0281 | -0.2415 | **1.0000** | -0.4127 | -0.0254 |
| $\acute{\theta}_y$ | 0.0658 | -0.2486 | -0.2312 | **0.1147** | 1.0000 | -0.4151 |
| $\acute{\theta}_z$ | -0.0277 | -0.0804 | 0.8290 | **0.0238** | 0.3619 | 1.0000 |

# 6. Conclusion

In this study, a thorough optimization problem was developed. A 6-DOF engine model was considered and the optimization process including objective function, optimization variables and constraints were described. The upper and lower bounds of variables, especially the position of mounts, were chosen carefully based on real limitations on the chassis for locating the mounts. Therefore, the minimization of objective function became more limited by forcing the variables to be within the bounds of the possible ranges. By considering limitations and constraints, feasibility of solution obtained from optimization was guaranteed. Although, the complete decoupling of modes was not possible due to the strong constraints in the system design, the results demonstrated that the noticeable improvement has been caused in mode decoupling of the engine. Comparing the optimal configuration with the original one, the results confirmed that the optimal design of engine mounting system has a significant effect in frequency response of the engine CG movement. Achieving more decoupled system, the optimal design has a better chance of placing dominant natural frequency blow the operation range. Also, the transmitted forces to the chassis were reduced noticeably by the mentioned optimization. This optimization problem can be adjusted for any specific case by applying new conditions.

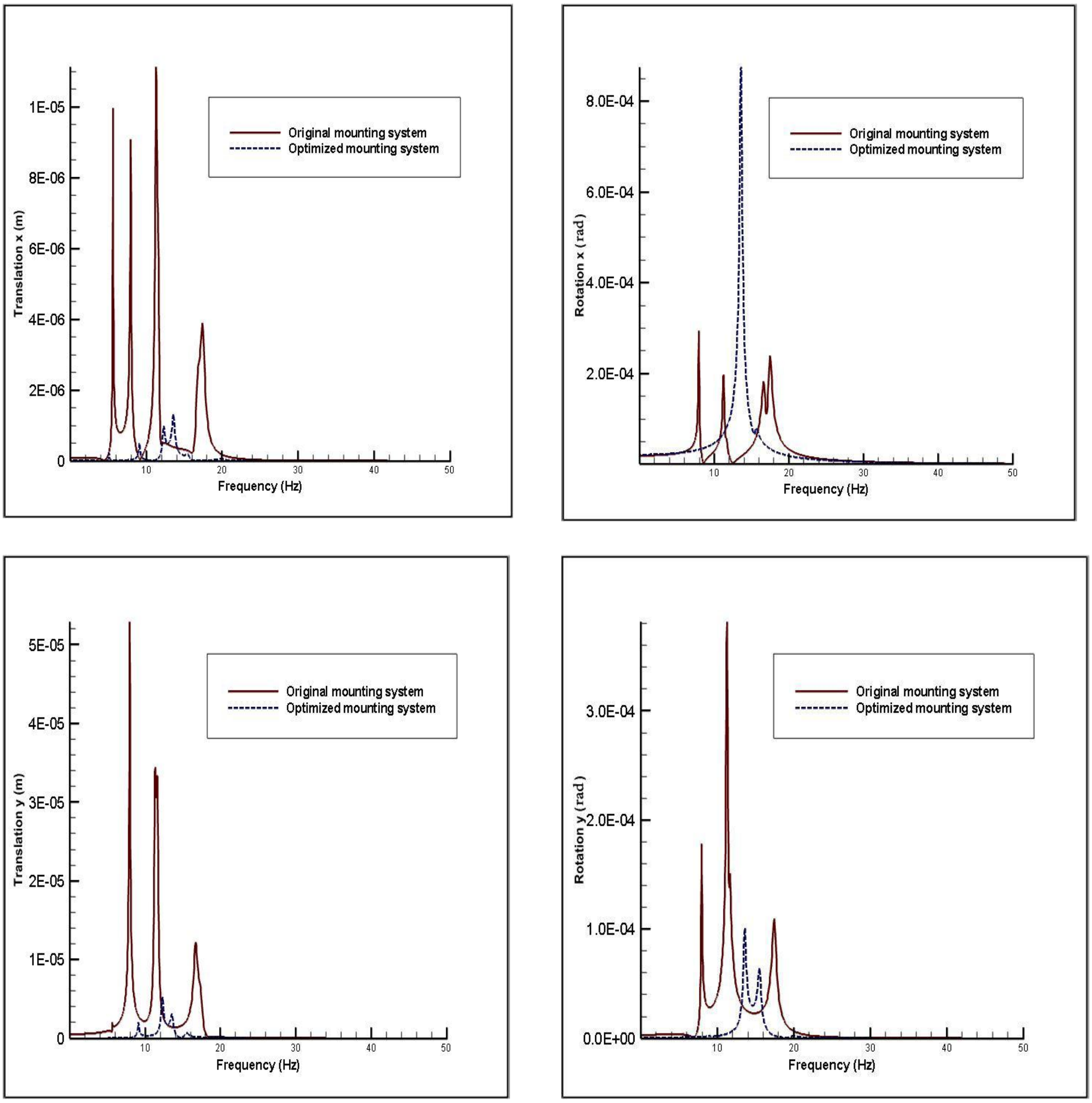



**Fig.3**. Frequency responses of CG movements in TRA coordinate system for original and optimal engine mounting system (for unit torque excitation along crankshaft)

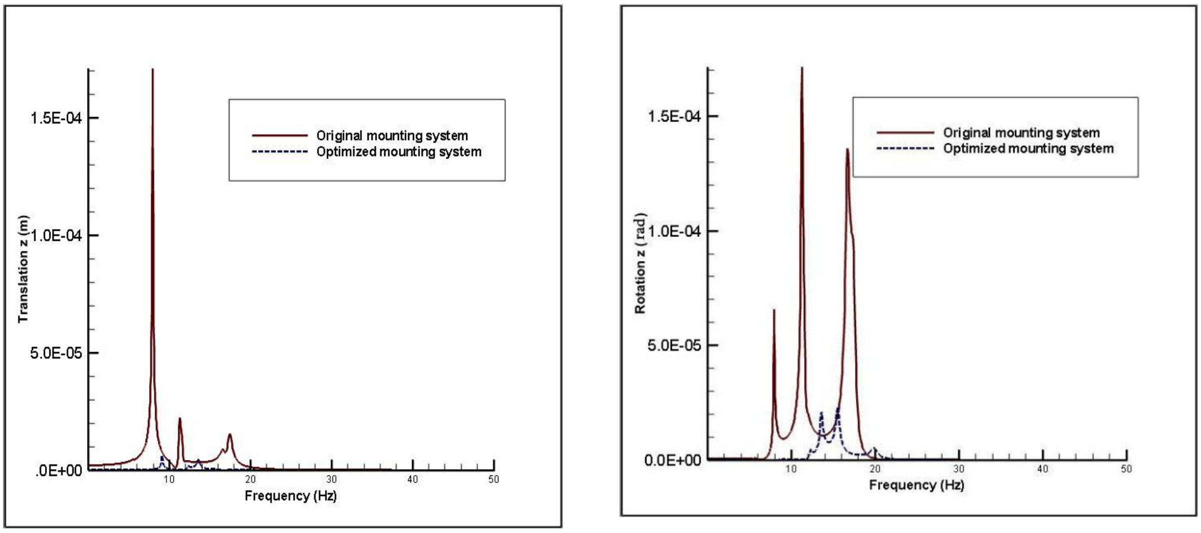
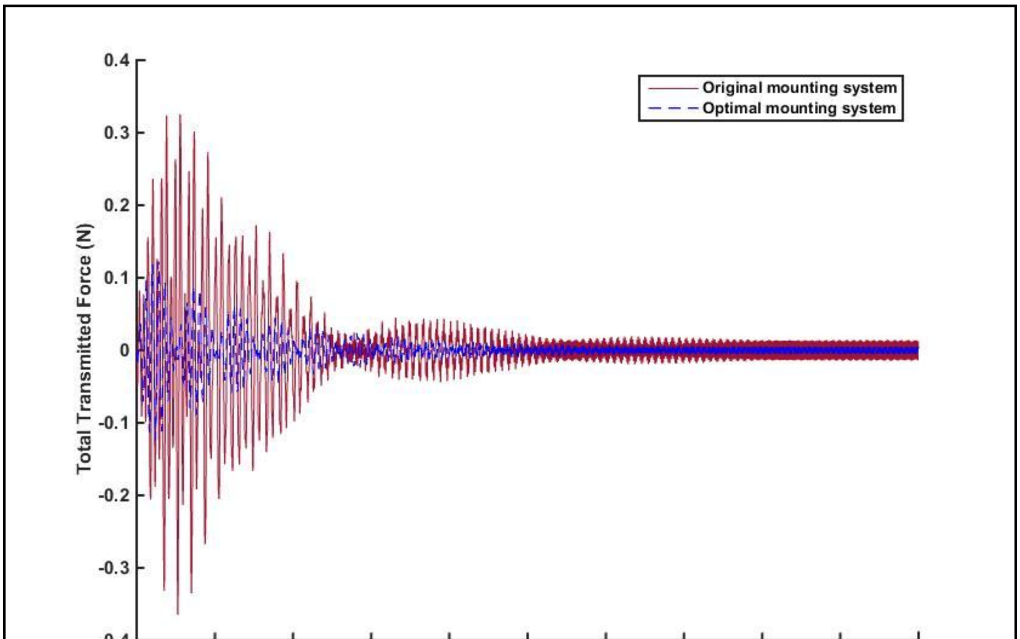
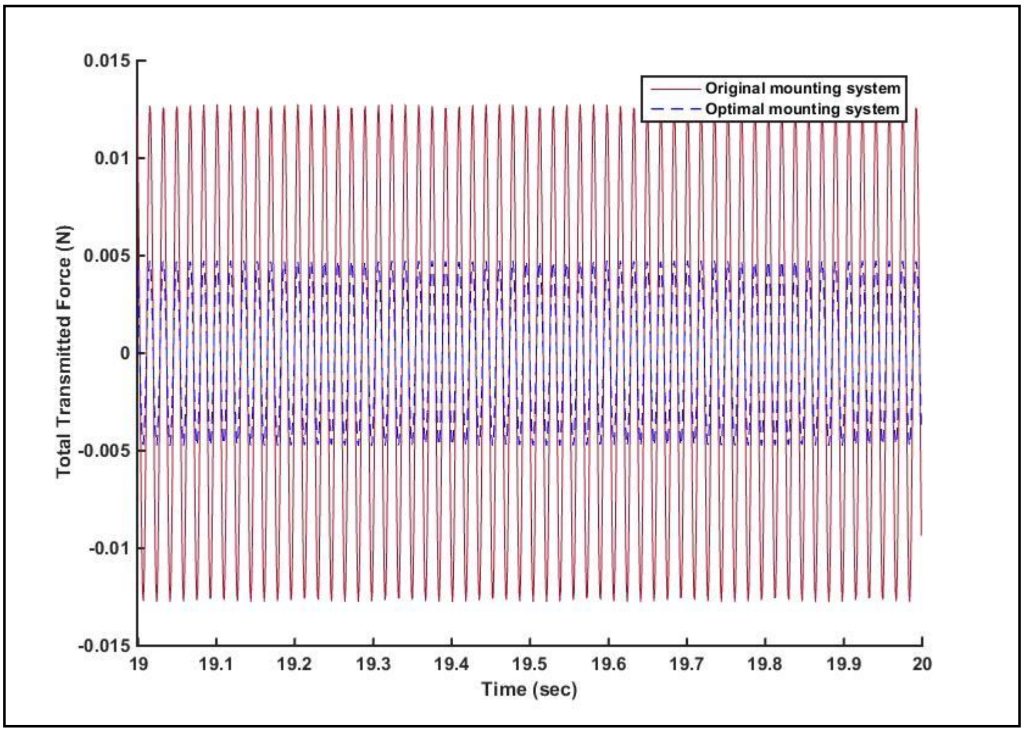



**Fig.3**. . (Continued)



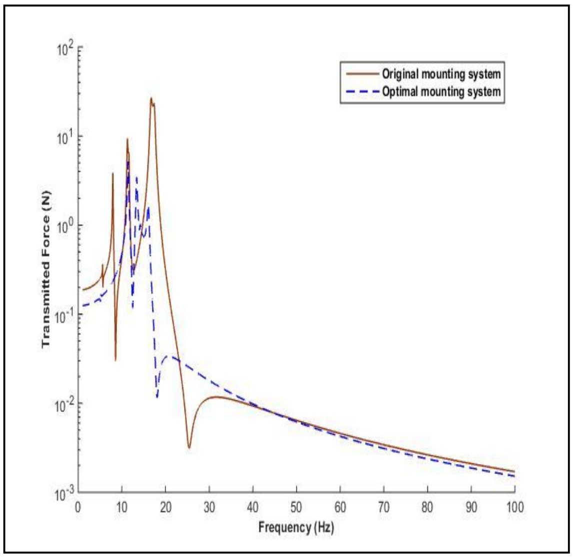

(a)

**Fig 4**. Time response of total transmitted force to chassis in z-direction for original and optimal engine mounting system (a) Transient response (b) Steady-state response

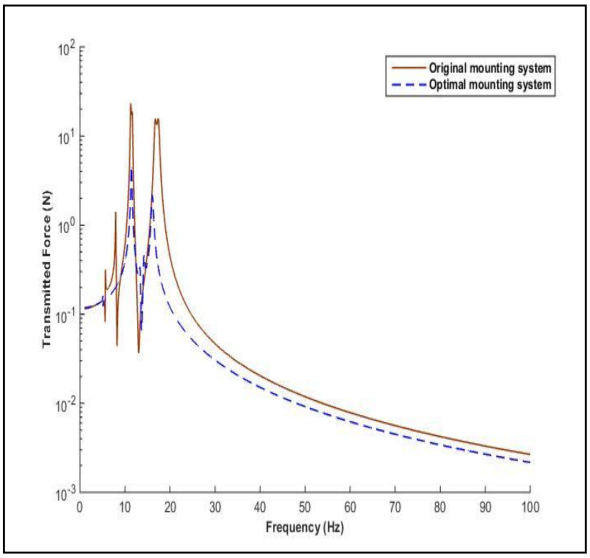



(b)

**Fig 4**. (Continued)

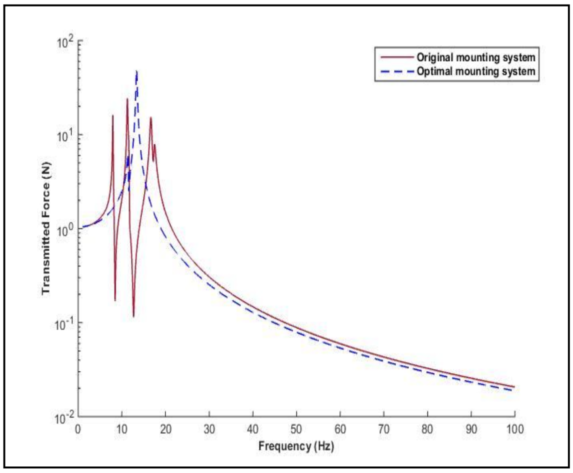



(a)

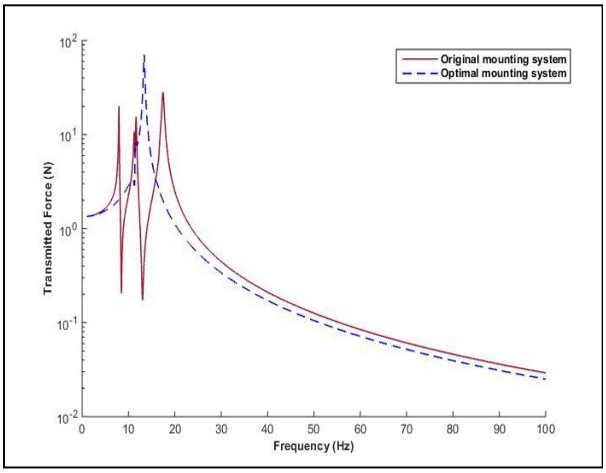



(b)

**Fig. 5**. Frequency responses of transmitted force to chassis in the position of (a) first (b) second (c) third and (d) fourth mount for the original and optimal engine mounting system (for unit torque excitation along crankshaft)




(c)




(d)

**Fig. 5**. (Continued)

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