

Custom Design Multi-Axial Engine Mount Load-Cell Development for Road Load Identification and Fatigue Life Estimation

Umud Esat Ozturk and Lutfi Ucar
Ford Otomotiv Sanayi A.Ş.

Kaveh Shahidi and Nuri Ersoy
Bogazici University

Onur Zobi and Umit Bagdat
Ford Otomotiv Sanayi A.Ş.

Rıfat Yanarocak
Design Verification & Test Engineer

Serkan Elmalı
Ford Otomotiv Sanayi A.Ş.

ABSTRACT

Internal combustion engines are attached on the vehicle body using engine mounts composed of two cast iron brackets and a rubber isolator connecting them. Engine mount road load identification during vehicle durability tests on proving ground is a critical task for engine mount development. Using standard multi-axial load-cell is not possible unless major design revisions on the vehicle body or engine block are done. Using wheel force transducers and vehicle dynamic simulation tools need extensive model tuning work to get accurate load information. Hence, a custom multi-axial load-cell design is preferred for the engine mount load identification of a BCar I4 engine. The developed load-cell engine mount bracket can be installed without doing any design changes on the vehicle. Design, durability analysis, instrumentation, calibration and vehicle installation of custom designed six degree of freedom multi-axial load cell have been performed. Strain and acceleration measurements have been conducted on vehicle durability test track. Data have been processed to obtain forces and moments in six degree of freedom on the engine mounts. The loading information was used for fatigue life estimation.

CITATION: Ozturk, U., Ucar, L., Shahidi, K., Ersoy, N. et al., "Custom Design Multi-Axial Engine Mount Load-Cell Development for Road Load Identification and Fatigue Life Estimation," *SAE Int. J. Mater. Manf.* 9(3):2016, doi:10.4271/2016-01-0413.

INTRODUCTION

Road load identification has always been a critical task in engine mount development program. Engine mounts are mounted in one end onto the car body and on the other end onto the engine. Any design changes on the powertrain, body and suspension change the load transmitted by the engine mounts. Rubber compounds used in engine mounts are frequently changed due to NVH and durability issues. Change in rubber stiffness also changes the dynamic behavior of engine assembly and the loads on the engine mounts.

A typical passenger car engine mount configuration is shown in Fig. 1. The left-hand-side (LHS) engine mount is attached onto the transmission and the right-hand-side (RHS) mount onto the front side of the engine block. There is a third engine mount called rollrestrictor (RR) to restrict the rolling motion of the engine assembly. LHS and RHS engine mount are subjected to multi-axial loading whereas the RR engine mount is mainly under uniaxial tension-compression

loading. There are many components around the engine mounts which are not shown in Fig. 1. Those components are placed at least 20 mm away from the engine mounts.

The common method for the identification of road loads on different components of the vehicle is to wheel force transducers (WFT) and multi-body dynamics simulations. Cruz et al. [1] showed how to incorporate WFT data measured at different test tracks into the ADAMS multi-body dynamics simulations to obtain load cases of different components.

This approach is convenient when WFT equipment and ADAMS models or the budget required to invest on them are available. Resources and expertise for building and tuning the multi-body dynamics models should also be considered because of the frequency dependent and nonlinear response of suspension system and other rubber components.

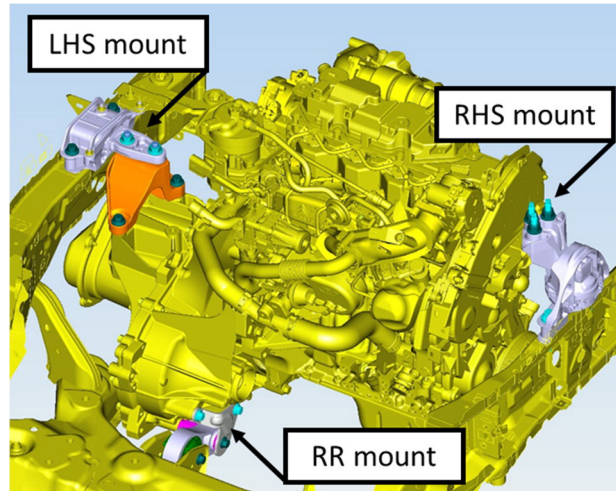


Figure 1. LHS, RHS and RR engine mounts mounted on B-Car body and I4 engine.

Another approach is to directly measure the engine mount loads using off-the-shelf multi-axial load transducers. In this case, there is packaging problem because of size of multi-axial force transducers. A multi-axial force transducer of a load range in z direction larger than 10kN has around 30cm diameter and 10cm height [2]. It is not possible to place them on a passenger car without doing major design changes on the body and engine block. On the other hand, direct measurement of loads are the most accurate and in some cases most cost effective option for the case where there is no WFT data and calibrated multi-body dynamics model.

Considering the packaging drawback of direct measurement option of engine mount loads, it is decided to develop six degree-of-freedom multi-axial load-cell brackets. In this paper, design, instrumentation, calibration, data acquisition and fatigue calculation steps will be presented for a B-Car I4 engine application.

SIX DOF MULTI-AXIAL FORCE SENSOR DEVELOPMENT

Force-Sensor Evaluation

Maltese cross or cruciform shape is the most common shape for 6 DOF multi-axial force sensor because it enables mechanical decoupling of different force components. Chao et al. [3] studied different strain gage configurations and their effect on the condition number of calibration matrix. Shi and Hall [4] developed a six-axis force sensor for measuring the loading of the racing tire because of similar limitations of using off-the-shelf WFT. Shahidi and Ersoy [5] used 12 strain gages to successfully decouple the force components on a Maltese cross type 6 DOF multi-axial force sensor. Ford Otosan Product Development team worked with Shahidi and Ersoy to successfully develop their force sensor for a B Car I4 engine mount presented in this study.

In this study, 12 strain gages were used to measure the six force components for the load-cell developed. The strain gage configuration is shown in Fig. 2. There are 3 strain gages at each

slender beam of the load-cell. Finite element analysis were performed to optimize the location of the strain gages and to predict the durability of the load-cell (Fig. 3).

There are six components of the input force vector $\{F\} = \{F_x, F_y, F_z, M_x, M_y, M_z\}$. In order to relate the input load vector $\{F\}$ to the output strain vector $\{E\}$, a transformation matrix $[S]$ must be calculated.

$$\{E\} = [S] \{F\} \quad (1)$$

Calibration matrix $[C]$ is the inverse of $[S]$,

$$\{F\} = [S]^{-1} \{E\} = [C] \{E\} \quad (2)$$

The entries of strain matrix $[S]$ can be calculated using Eqn. 3. ε_j ($j = 1, 2, \dots, 12$) are the strain values measured by strain gages and E_{ij} ($i = 1, 2, \dots, 6$ and $j = 1, 2, \dots, 6$) are the entries of $[S]$. They are obtained by using ε_j values at 12 different locations induced by corresponding external loading during calibration measurements.

$$E_{1i} = (\varepsilon_1 + \varepsilon_5) - (\varepsilon_3 + \varepsilon_7) \quad (3a)$$

$$E_{2i} = (\varepsilon_4 + \varepsilon_8) - (\varepsilon_2 + \varepsilon_6) \quad (3b)$$

$$E_{3i} = (\varepsilon_1 + \varepsilon_2 + \varepsilon_3 + \varepsilon_4) - (\varepsilon_5 + \varepsilon_6 + \varepsilon_7 + \varepsilon_8) \quad (3c)$$

$$E_{4i} = (\varepsilon_2 - \varepsilon_4) - (\varepsilon_6 + \varepsilon_8) \quad (3d)$$

$$E_{5i} = (\varepsilon_1 - \varepsilon_3) - (\varepsilon_5 - \varepsilon_7) \quad (3e)$$

$$E_{6i} = (\varepsilon_9 - \varepsilon_{10}) - (\varepsilon_{11} - \varepsilon_{12}) \quad (3f)$$

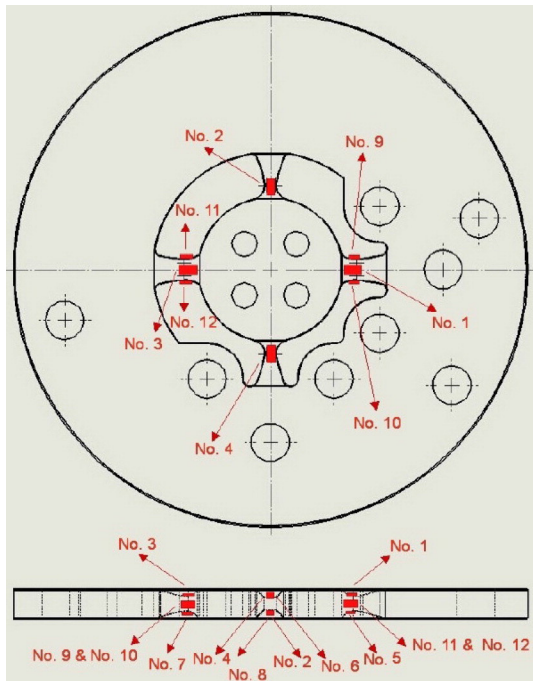


Figure 2. Location of strain gages.

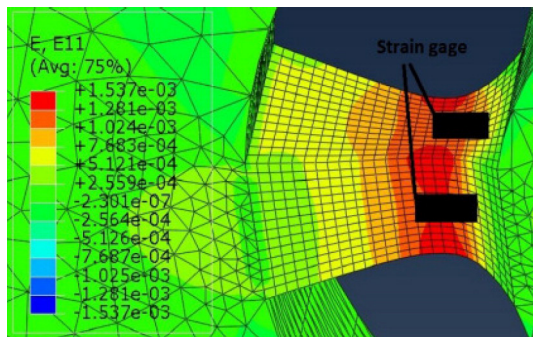


Figure 3. Strain calculation at the strain gage locations by finite element analysis.

Calibration

Special tooling was designed and manufactured for the universal tensile testing machine to apply six force components onto the load-cell. For translational loadings, actuator and load-cell of the universal tensile testing machine were used (Fig. 4). Weights suspended using a moment arm to apply moments (Fig. 5). Forces and moments plotted against the measured strain values in Figs 6, 7, 8, 9, 10, 11. The linearity and hysteresis of the force vs. strain curves were satisfactory within the expected force ranges. Slopes of each force vs. strain line were used in Eqn. 3 to obtain the entries of $[S]$.

In Table 1, $[S]$ entries are given for the following load and moment magnitudes: $F_x = F_y = F_z = 1 \text{ kN}$ and $M_x = M_y = M_z = 1 \text{ Nm}$. Calibration matrix $[C]$ was obtained by taking inverse of $[S]$. The calibration matrix is a diagonal one, which means low cross talking will be seen among different force components. Mechanical decoupling was successfully achieved using the Maltese cross shape.

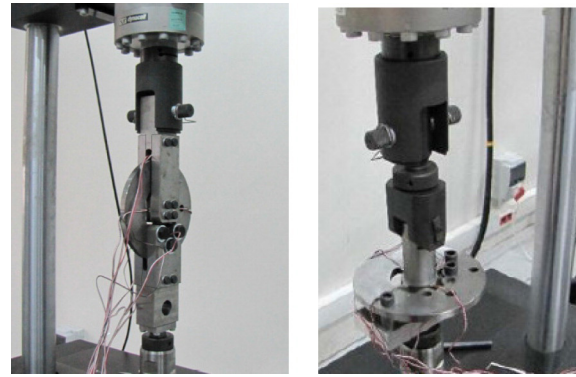


Figure 4. Test setup for applying translational forces during calibration test.

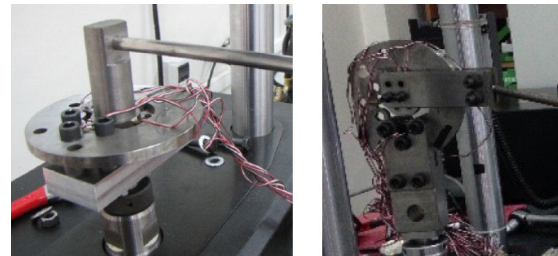
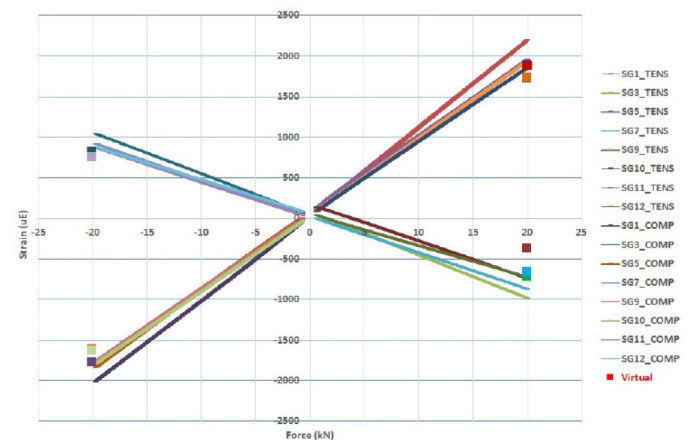
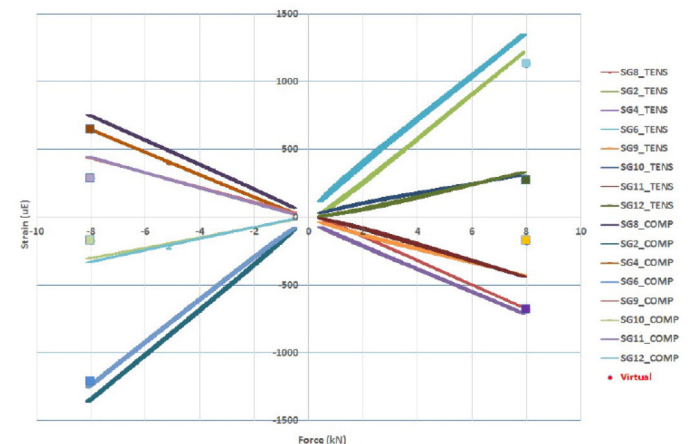


Figure 5. Test setup for applying moments during calibration tests.

Figure 6. F_x vs ϵ_i ($i=1, \dots, 12$) curves obtained during calibration tests.Figure 7. F_y vs ϵ_i ($i=1, \dots, 12$) curves obtained during calibration tests.