# Design and Analysis of Clutch Housing Prototype for One Cylinder Diesel Engine Direct Clutch System

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***Abstract***

In this work, a design of clutch housing for one cylinder diesel engine in an agricultural truck or E-TAND using CAD and CAE was carried out. A CAD model was analyzed by using finite element method ANSYS® software. The analyses were consisted of structural stress and fatigue life analysis by converting vibration profile into corresponding forces which were used as a load parameter. In order to collect relevant load profiles, an experiment was set up to measure vibration amplitude by the engine using accelerometers. Moreover, an optimization technique was used to find an optimum design of a clutch housing suitable for one cylinder diesel engine application. Overall design frame work, concerning design problem, load collecting experiment, and computational analysis results were reported and discussed. After achieving the goal of design, all parts will be ready for prototype manufacturing and assembly for a direct clutch system in an agricultural truck.

**Key words:** Finite element analysis, Stress analysis, Design optimization

## 1. Introduction

An agricultural truck is used widely in rural parts of Thailand. The most basic and the most famous type is E-TAND because of its relatively low manufacturing and maintenance cost. E-TAND is also known as a multipurpose vehicle in some regions. A one cylinder diesel engine is generally used as a prime mover from which the power will be transferred to gearbox via belts. V-belt type is generally selected. However this type of transmission is accompanied by considerable amount of loss. In order to develop an improved transmission system for E-TAND, a direct clutch system was designed for one cylinder diesel engine. Nevertheless, one cylinder diesel engine was not originally designed for such clutch system. Thus, there were several parts that were needed to be designed in order to connect a clutch directly to a flywheel such as clutch contact surface, clutch housing, and clutch fork. In this work, a design of clutch housing was focused using finite element analysis and optimization tool. All design processes involved in this work are summarized in Fig.1

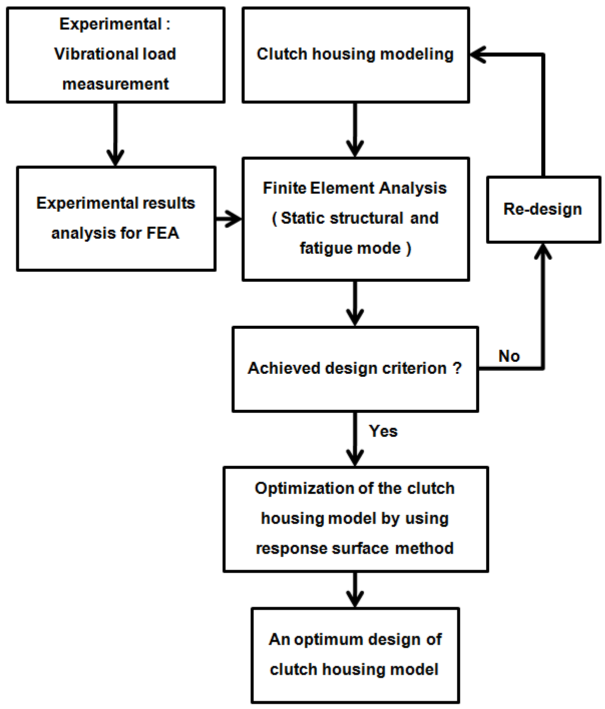


Fig. 1 Clutch housing design flowchart

The finite element method is used to approximate a problem of stress analysis, fluid flow, heat transfer, etc. In recent years, there were many studies using the finite element method to predict failures or analyze resulting stresses. In automotive industry, this method was used to analyze failure or fatigue life of various automotive parts such as universal joint yoke and a drive shaft [1], fatigue life prediction of lower suspension arm [2], rear axle housing [3]. These authors employed a similar kind of sequence or research methodology which involved modeling a CAD model, performing stress and fatigue life analysis in which the boundary conditions and loads were needed, and validating results by comparison with experimental results. In some cases, after the finite element method was used to analyze a stress, an optimization process was carried out to determine a suitable redesign for structure. Various algorithms has been investigated in literature. For example, N. Kaya et al. used the design of experiment (DOE) to optimize a clutch fork using topology and shape optimization by the response surface method [4]. In addition, an optimization algorithm using orthogonal arrays in discrete design space for structures was carried out by Kwon-Hee Lee et al.[5].

## 2. Clutch Housing Design

### 2.1 Specific concerns in clutch housing design for one cylinder diesel engine

As shown in Fig.2 clutch housing is a part located between engine and gearbox that covers all of the clutch system. The design was constrained by an access point of clutch fork, dimension of clutch system, and flange of gearbox. The special concerning was a potential failure caused by the vibration of engine. In addition, an alignment problem between gearbox and engine was considered.

### 2.2 Clutch housing modeling

Clutch housing was designed according to the dimension of clutch set and a mounting location. According to its characteristic, one cylinder diesel engine is a high vibration engine. Therefore, a mounting plate was designed to avoid a misalignment between gearbox and engine when acceleration was required. Components of clutch housing prototype model are shown in Fig.3.

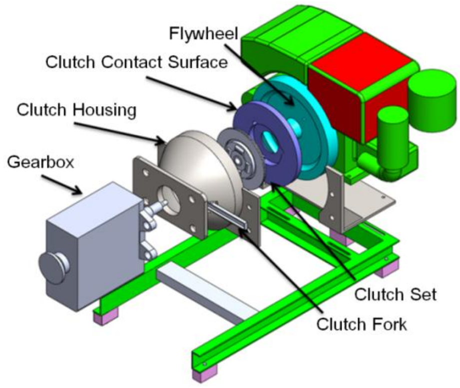


Fig. 2 A layout of direct clutch system prototype

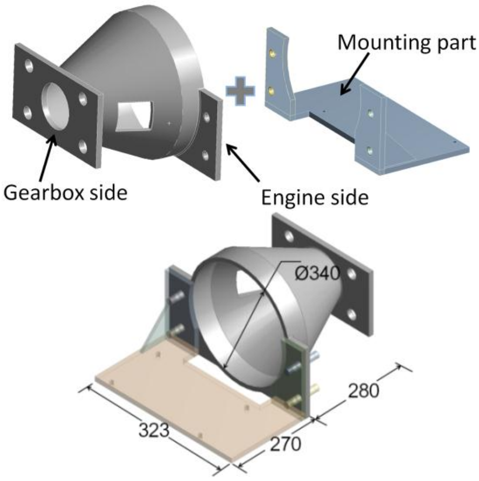


Fig. 3 A clutch housing prototype model (in mm.)

## 3. Experimental - Engine Vibration Profile Measurement

In present work, vibrational load profiles measurement at various engine load conditions was carried out on a dynamometer test configuration.

### 3.1 Experimental apparatus

The apparatus of this experiment(Fig. 4) were consisted of

- 14HP,Direct injection engine (KUBOTA RT140)
- Data acquisition tools :NIcDAQ-9172 and NI 9234
- Accelerometers; Brüel&Kjær Type 4507 B002
- Dynamometer with Dynamo controller
- NI Sound and Vibration Assistant software

### 3.2 Experimental setup

In this experiment, the engine was operated at 3 different load cases which were idle load(1120 rpm), partial load (60% of throttle, 1800 rpm), and full load (100% of throttle, 2200 rpm). The accelerometers were connected with NI DAQ set and were attached at front and rear end of engine wood base plate as shown in Fig. 5. The results were recorded by using NI Sound and Vibration Assistant software with a sampling rate of 1000Hz. The experiments were repeated for 3 times at each location for all load cases to confirm a repeatability of the results.

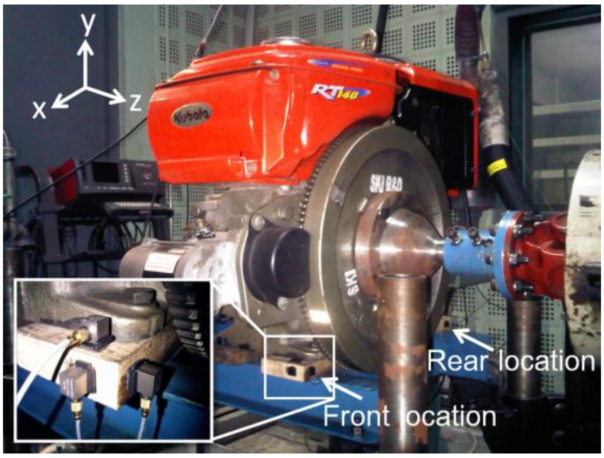


Fig. 5 Accelerometers attachment configuration near engine base

Fig. 4 An experimental set up for vibration load measurement: (from left) DAQ equipments and accelerometer, dynamometer load controller, and dynamometer chassis with test engine.

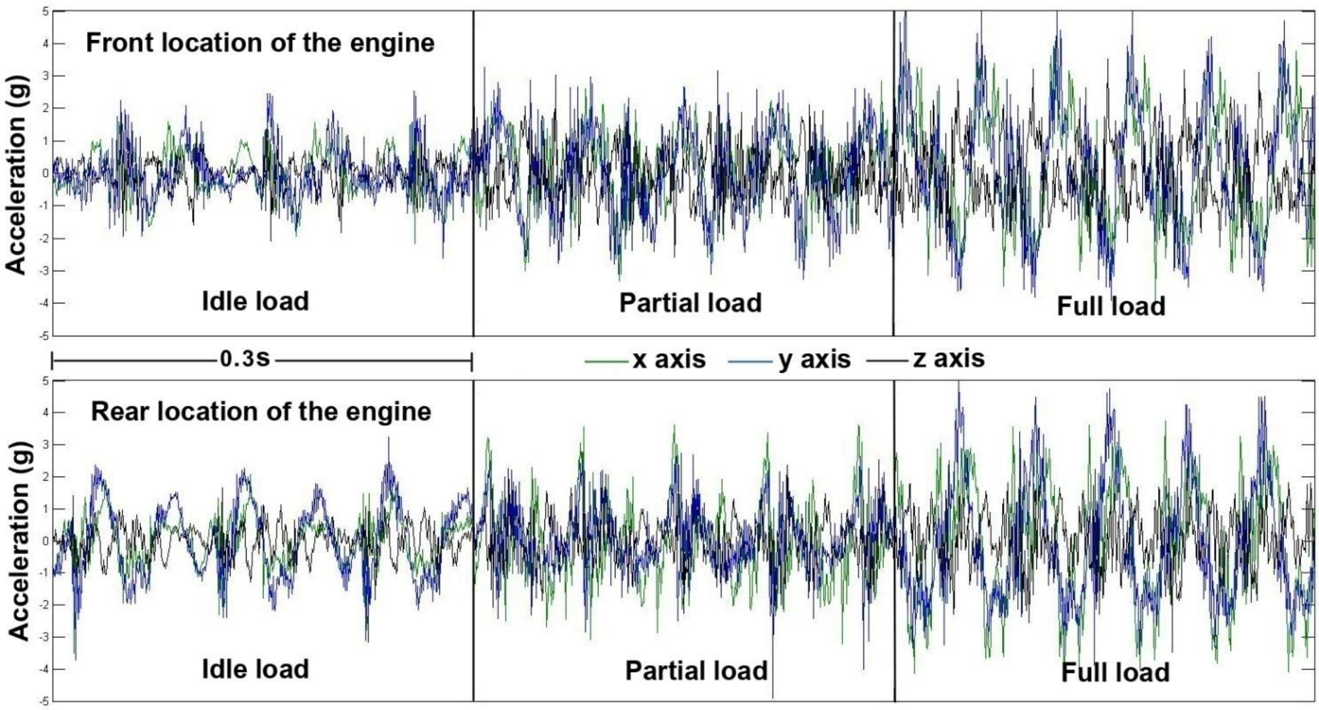


Fig. 6 The engine vibration profiles measured at various loads.

### 3.3 Experimental results

The results were separated into three main cases with two measurement sensor locations for each case. The results are shown in Fig. 6. Generally, all measured responses displayed a sinusoidal pattern for all load cases. Highest vibration amplitudes were observed during full load conditions in x and y axis directions.

In order to use the experimental data effectively in simulation, the data were passed through a low pass filter to collect corresponding data in frequency range under 60Hz. A graphical comparison between raw data and filtered data is shown in Fig. 7.

Additionally, in order to carry out a structural and optimization analysis, an input parameter should be applied as a force. Therefore, the measured acceleration was converted into corresponding force by multiplying with an engine effective mass. Assuming that C.G. of engine was located at center of a flywheel, effective mass of engine at front and rear support locations were calculated to be 73.66 kg and 38.44kg.

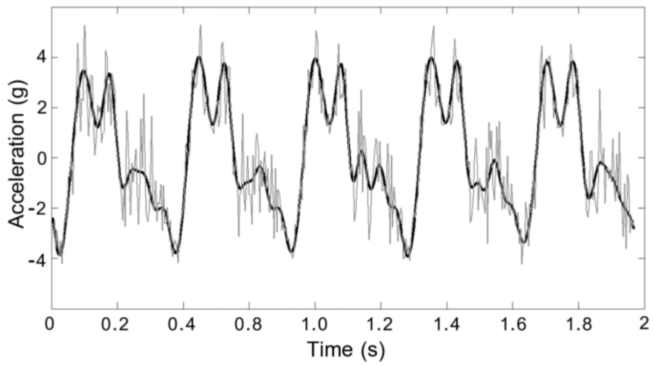


Fig. 7 Comparison between filtered and non-filtered vibration data

The calculated maximum forces in each load case which were used in simulation can be summarized in Table. 1. It can be seen that the maximum force under full load condition was higher than those under partial and no load conditions by approximately a factor of 1.89 and 2.84 respectively.

Table. 1 Maximum force in each load case

| Cases | Maximum Force (N) | | |
|---|---|---|---|
| | X | Y | Z |
| Front no load | 1232 | 1564 | 792 |
| Front partial load | 1590 | 1640 | 1348 |
| Front full load | 2872 | 3048 | 2620 |
| Rear no load | 852 | 634 | 360 |
| Rear partial load | 1224 | 916 | 712 |
| Rear full load | 2346 | 1686 | 1446 |

Furthermore, for fatigue failure mode simulation, a load history was required. As a results, vibration profiles from each experiment cases were combined together with reference to TIS (Thai Industrial Standard) 2155-2546 driving cycle which is an equivalent of EUDC (European Driving Cycle Test) shown in Fig. 8. The assumption of a relationship between a driving cycle and an engine load were defined such that while the vehicle was accelerate, the engine ran with full load. On the other hand, the engine ran at partial load and no load, when vehicle ran on cruise speed and decelerated respectively. The loading history which was used in this study can be seen in Fig. 9.

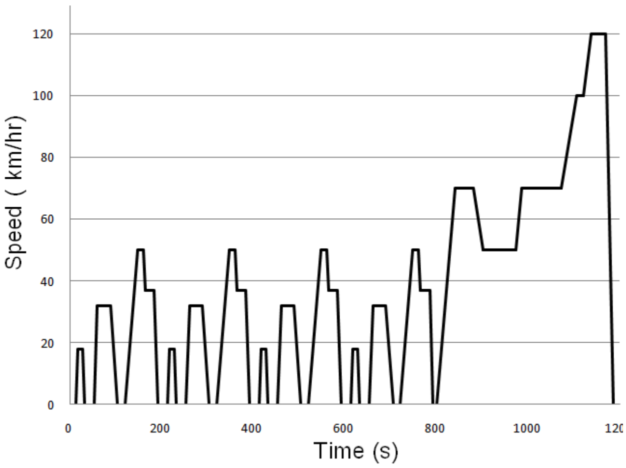


Fig. 8 EUDC standard: Driving cycle for 1 day,[6]

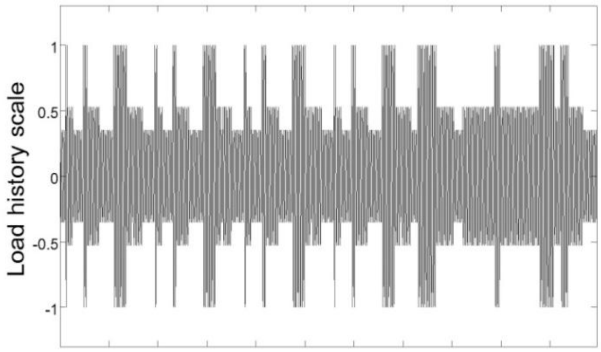


Fig. 9 Combined loading history for fatigue analysis

## 4. Finite Element Analysis

In order to determine the stress generated and analyze a fatigue failure of the clutch housing, the finite element analysis was applied on to the CAD model.

### 4.1 Finite element modeling

Mesh model of assembled clutch housing was prepared using ANSYS® software. Two types of mesh element were used, i.e. hexahedron for bolts and mounting part, and tetrahedron for clutch housing with a global element size of 6mm, 4mm, and 5mm respectively. In addition, each part was contained with elements which had skewness value lower than 0.75. The clutch housing FE model is illustrated in Fig. 10.

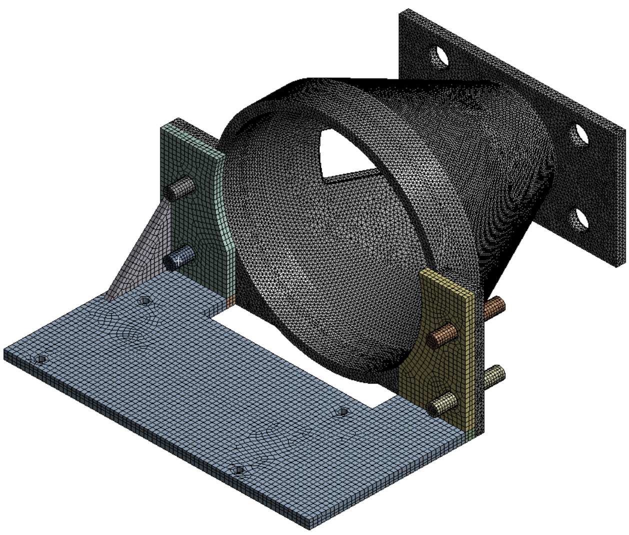

Fig. 10 Finite element model of clutch housing

### 4.2 Computational setup

For all computational analysis, a structural steel was chosen as material for all parts. Corresponding mechanical properties were taken from engineering data in ANSYS® software, including S-N curve for fatigue analysis. For an initial simulation, all components were assigned a thickness of 13mm with Clutch housing slope angle of 65 degree.

Boundary conditions applied in this study are shown in Fig. 11. The clutch housing was considered under a stress when loaded by engine vibration. The loads collected from the experiment were applied as a force loading. An amplitude of force loading was referred to maximum force in full load vibration (Table. 1). In addition, the input force loading was applied on both front and rear holes at the engine base plate of mounting part. A amplitude force was used in a static structural analysis while combined force amplitudes (Fig. 9) were applied as a load history for fatigue analysis. On the other hand, a fixed support was applied on the inner holes surface flange of clutch housing at the gearbox side. Moreover, the mean stress Goodman criterion was used in this work.

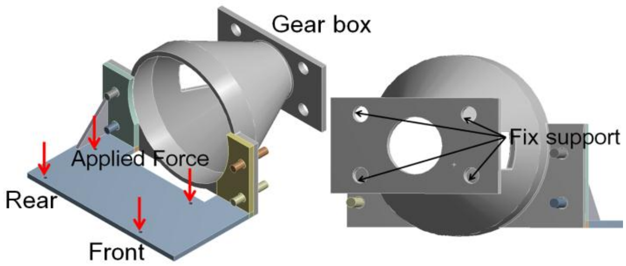


Fig. 11 Boundary conditions for clutch housing analysis

## 5. Size Optimization by Response Surface Method

ANSYS® Design Exploration Response Surface with the full 2nd order polynomial algorithm was used in this study to optimize the clutch housing prototype model. The thickness of clutch housing (DV1) and slope angle of clutch housing (DV2) shown in Fig. 12 were set as variable parameters. A relationship between maximum von-Mises stress and component geometries was studied during the optimization process by varying the clutch housing thickness was varied from 7mm to 13mm and the slope angle of clutch housing from 64 degree to 67 degree. The goals of this optimization were to reach a safety factor target of 3, to minimize volume of clutch housing, and to have no failure in fatigue mode. The design of experiment (DOE) used 3 factor levels and 2 parameters ($3^n$ factorial design) i.e. $3^2$ = 9 design points. Each parameter was divided into 3 ranges of value, maximum, middle, and minimum for each factor design point. After 9 design points were completed, a response surface model was plotted using the full 2nd order polynomial.

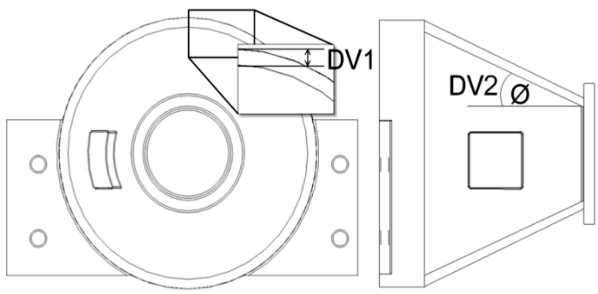


Fig. 12 Design variable of clutch housing optimization, thickness (DV1), slope angle (DV2)

## 6. Results & Discussions

Initially, the analysis was carried out with a 13 mm thickness and 65 degree of slope angle. The resulting stress is shown in Fig. 13.

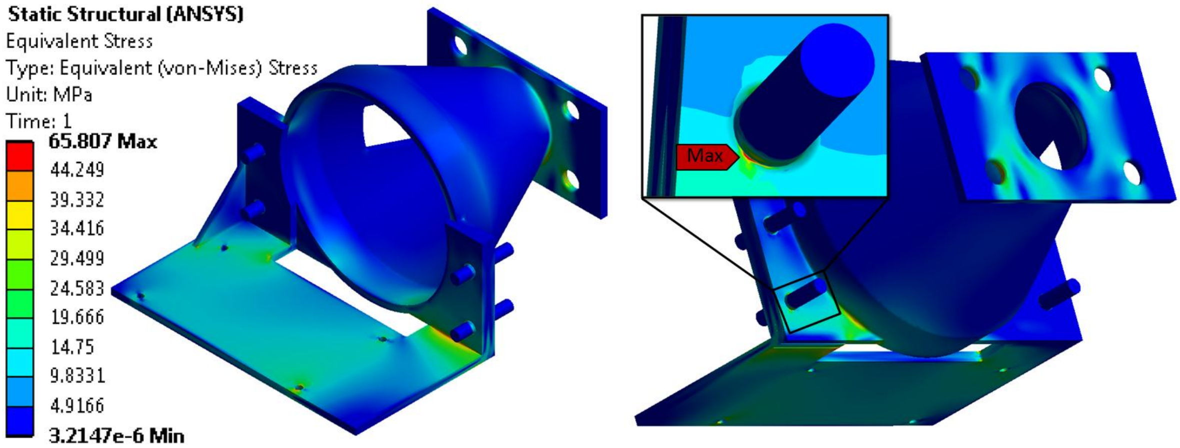


Fig. 13 von-Mises stress distribution in clutch housing, component thickness of 13 mm and 65 degree of slope angle.

Maximum von-Mises stress was 65.807 MPa which gave a safety factor of 3.799 with non-failure of fatigue mode. Since the safety factor was higher than a target value of 3, indicating overdesign, the method of optimization was applied to find an appropriate thickness and slope angle of clutch housing to achieve the design goals. During optimization, the DOE was carried out as explian in previous section and the calculated results are shown in Table. 2.

Table. 2 DOE- $3^2$ design table results

| Run # | DV1 (mm) | DV2 (degree°) | Maximum von-Mises Stress (MPa) | Volume ( $x10^3$ $mm^3$) | Minimum Life (Days) |
|---|---|---|---|---|---|
| 1 | 7 | 65.5 | 116.45 | 5063.19 | 188.22 |
| 2 | 7 | 67 | 112.02 | 5099.74 | 260.05 |
| 3 | 7 | 64 | 123.40 | 5025.50 | 121.64 |
| 4 | 10 | 65.5 | 81.05 | 5784.78 | 27563 |
| 5 | 10 | 67 | 63.61 | 5836.92 | 27563 |
| 6 | 10 | 64 | 85.06 | 5731.12 | 27563 |
| 7 | 13 | 65.5 | 63.93 | 6523.23 | 27563 |
| 8 | 13 | 67 | 58.54 | 6590.96 | 27563 |
| 9 | 13 | 64 | 69.82 | 6453.60 | 27563 |

Based on the DOE results in Table. 2, the relationship between DV1, DV2, and maximum von-Mises stress as well as volume was constructed to form a response surface by using 2nd order polynomial algorithm. The equations of relationship are shown in Eqs. (1) and (2) where x is DV1 and y is DV2.

Max. von-Mises stress (MPa) = $-2660.323 - 40.6072x + 96.0588y + 1.5688x^2 - 0.7712y^2 + 0.0056xy$

Eq.1

Volume ($x10^3 mm^3$) = $1431.32 - 7.3375x + 61.6688y + 0.9426x^2 - 0.47134y^2 + 3.541xy$

Eq.2

The corresponding response surface of clutch housing design parameters for maximum von-Mises stress is displayed graphically in Fig.14.

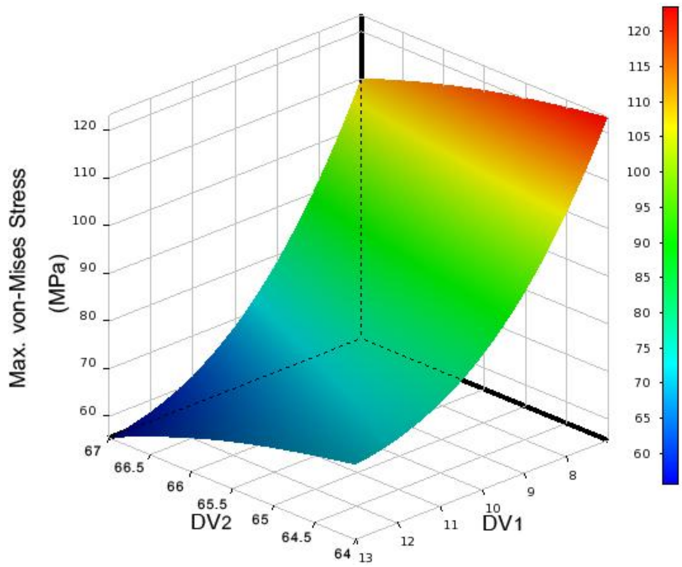


Fig. 14 Maximum von-Mises stress response surface for clutch housing design parameters (DV1,DV2).

According to the response surface, a sensitivity of the clutch housing slope angle was relatively low since incresing of slope angle resulting in a slight decrease in a maximum von-Mises stress. In contrast the variation of clutch housing thickness showed very high sensitivity such that maximum von-Mises stress significantly increased while the thickness was decreased. In addition, increasing of both parameters would result in an increase of the total volume of the model.

In this optimization, one of the goals was to minimise volume while a safety factor had to be higher than 3 or a maximum von-Mises stress of 83.33 MPa. Hence, 83.33 MPa was used as max. von-Mises stress in Eq. (1), to calculate possible x and y. Moreover, every possible combinations of x and y results were substituted into Eq. (2) to determine the minimum volume. From this calculation, the optimum point result was 8.7 mm of thickness and 67degree of clutch housing slope angle.

In addition, this optimum point was used in analysis to validate the results. Maximum von-Mises stress was 84.74MPa with a corresponding safety factor of 2.95 and structural volume of 5515.43 $x10^3$mm.$^3$, while fatigue life was also achieved. The percentage error of maximum von-Mises stress, safety factor, and volume was 1.67%, 1.69%, and 0.06% respectively which were in acceptable ranges. Therefore, 8.7 mm. of thickness and 67degree of slope angle were optimum point of this clutch housing

Table. 3 Results comparison of optimum design point from response surface method

| | Calculation | Simulation | Error(%) |
|---|---|---|---|
| Max. von-Mises stress (Mpa) | 83.33 | 84.74 | 1.67 |
| Safety factor | 3 | 2.95 | 1.69 |
| Volume ($x10^3mm^3$) | 5518.84 | 5515.43 | 0.06 |
| Fatigue life | Pass | | |

## 7. Conclusion

The present work involved designing and analyzing a clutch housing for a direct clutch system prototype for a one-cylinder diesel engine used in agricultural truck or E-TAND. The specific design concerns were discussed and details of vibration loading measurement, finite element analysis, and optimization of a clutch housing were explained.

In vibration loading measurement, an engine vibration loading was measured and was later applied as input parameters (load) in a finite element analysis. The experiment showed that vibration loading amplitude of one cylinder diesel engine could increase from 0.002g to 4.22g with increasing engine load. The dominant vibration amplitude occurred on a plane corresponding to piston cylinder movement. Also, in order to use measured data effectively, a low pass filtering was needed.

For finite element method and optimization method, these methods were used to show the result of clutch housing behavior when under load. According to the response surface calculation, the sensitivity of both parameters that affect maximum von-Mises stress and volume of the clutch housing were presented. An increase of thickness significantly decreased a maximum von-Mises stress while increased a volume. In contrast, a variation of slope angle was accompanied by only slight effect. Therefore, the main concerned parameter was a thickness of clutch housing. Furthermore, the optimum points were founded at thickness of 8.7 mm. and 67 degree of slope angle with a coressponding maximum von-Mises stress and volume of 84.74 MPa and 5515.43 $x10^3 mm.^3$, respectively. In addition, results of current study display obvious advantages of the employed approaches such that;

- The sensitivity of each parameter can be shown and compared.
- Saving a computational time and shorten a design process.
- The optimum point prediction gave very accurate result. The error was not exceeding 1.7 %.

For future work, this optimum clutch prototype model will be manufactured and assembly into direct clutch system in agricultural or E-TAND for flied test and failure investigated.

## 8. Acknowledgement

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