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Research Article

DESIGN AND SIMULATION OF SUGARCANE LOADER STRUCTURE

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ABSTRACT:
The Entrepreneurs in small and medium size enterprises (SME) have an experience in sugarcane loader structure fabrication for a long time but it has not been developed with sufficient and appropriate engineering technologies. Thus, the purpose of this research is to design and analyze the strength of the sugarcane loader structure bases on engineering. The sugarcane loader structure was designed by Computer Aided Design (CAD) and Computer Aided Engineering (CAE), for the 45 horsepower KUBOTA ZL-1 Reverse tractor, which can install grab equipment to lift 300 kilogram of sugarcane mass up to 5 meters from ground, that total weight is approximately 3.5 tons. Using Finite Element Method (FEM) to simulate stress contour to occurs on the part with the sufficient structural strength to operate on the farm. The knowledge is gained from design and development of sugarcane loader structure bases on engineering which can be transferred to the entrepreneur for sugarcane loader structure proceeding. The 3D model of sugarcane loader structure was designed and analyzed under static condition with maximum hydraulic force. Moreover, the stress result of Finite Element Analysis was verified by using classical theory.

Keywords: *Sugarcane loader structure, Finite element method, Computer aided design***,** *Computer aided engineering*

1. Introduction

Thailand is one of the top leads of ASEAN in exporting sugar and sugarcane, there were 2.73 million tons had been exported to the region. As the market requires around 2.9 million tons of importing products including the announcement of AEC, these could be great opportunities for Thailand to expand the market within the region. In order to be competitive in the market, Thailand must focus on quality development as well as increasing the efficiency in producing as to decrease the costing. And as to succeed in those targets, many entrepreneurs have imported machines from overseas such as sugarcane loader to decrease transferring time of harvested sugarcanes to the factory and also decreasing the costing of permanent labors. But the prices of imported machines are quite high as well as the allowance for maintenance which are responsibilities for the entrepreneurs, therefore, there are some entrepreneurs have tried to developed the sugarcane loader by themselves by modifying the existing tractors with extra parts and material that can be found within the country as to be able to attach sugarcane loader structure Although, Thai entrepreneurs have experienced in sugarcane loader structure development for a long time but they have not been developed with sufficient and appropriate engineering technologies. Sugarcane loader structure is often failed when it is operated in farm. This means the equipment is not safe and also very dangerous for farmers. Therefore, the purpose of this research is to design and development sugarcane loader structure that bases on engineering knowledge which can withstand the load on severe condition, to study on Static Finite Element Analysis (FEA) of sugarcane loader structure, and to provide design procedure for Thai entrepreneurs.

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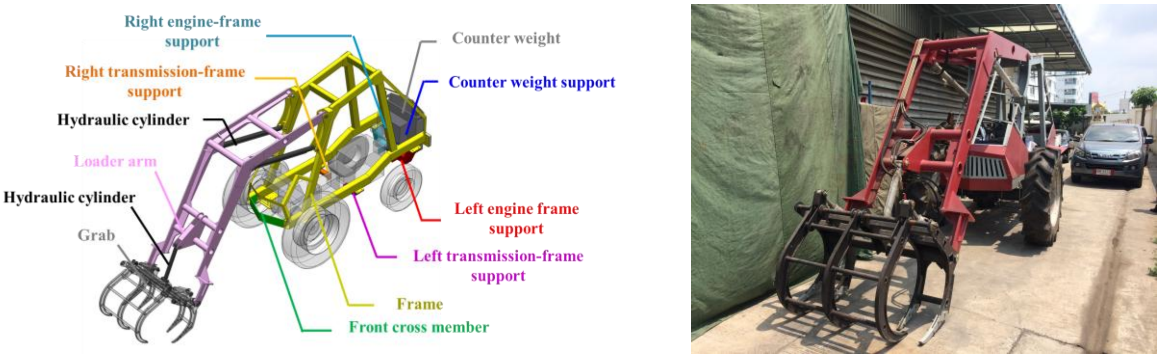
2. CONCEPT DESIGN

The concept designs were gathered from problems, recommendation about the producing and operating of the sugarcane loader structure. It could define the main design concepts as follow: first, the sugarcane loader structure could be able to detach and install to the tractor chassis. Second, Use KUBOTA SUNSHINE ZL-1-455 reverse tractor as the base of the sugarcane loader structure which the advantage of this model is the steering wheel and the seat can be swap with each other. The power is generated by 2.4 L of Diesel engine with 45 HP. The transmission can be switched from 2WD to 4WD. The suspension is rigid suspension (no spring and no absorber). The total tractor weight is 1.7 tons. The wheel-base is 1,980 mm and the wheel track is 1,450 mm. Third, the requirement of maximum load of sugarcane bundle is 300 kg. And fourth, the requirement of maximum height to lift the sugarcane bundle from the ground is 5 m.

3. GEOMETRY DESIGN

3D models are created by SolidWorks program. The 3D models are consisted with 4 main parts: loader structure, loader arm, counter weight and grab as shown in Fig. 1 (a).



(a) (b)

Fig. 1. (a) Geometry Design, (b) Prototype.

3.1 Loader structure

According to design concepts, the sugarcane loader structure must be able to mount on KUBOTA SUNSHINE ZL1-455 tractor. Thus, the loader structure is designed to be 8 components as follow: 1) Front cross member, the front cross member is a bending steel plate which was jointed ladders each side together. 2) Frame, the frame are consist of four rectangular steel columns, two rectangular steel ladders and two upper cross member that be fixed on rear axle and lower frame support. Cross section of columns and ladders are 75x100x6 mm and 100x150x6 mm respectively. Both side of loader arm and corresponding lift cylinders supposed to be mounted on frame. 3) Right engine-frame support, the right engine-frame support is lower support of frame which mounted to right side of the engine and also connected to counter weight support at rear of frame. 4) Left engine-frame support, the left engine-frame support function is similar to right engine-frame support but the shape is not symmetry. 5) Right transmission-frame support, the right transmission-frame support is lower support of frame which mounted to right side of the transmission. 6) Left transmission-frame support, the left transmission-frame support function is similar to right transmission-frame support but the shape is not symmetry. 7) Counterweight belt, Cause of vehicle stability, the counter weight was assembled to sugarcane loader. The belt is designed to mount balance weight with frame together and to make sure balance weight will not translate while loader arm operate in unstable farm ground. 8) Counter weight support, Counter weight support is designed to support counter weight around 1,100 kg that mounted to left and right engine -frame supports and engine. Each component is connected by bolts and nuts.

3.2 Loader arm

Normally, loader arm is designed as single arm which obscures the visibility of the driver. So to solve this problem, it is designed as a symmetrical double arm.

3.3 Counter weight

The design of the counter weight is to be the most harmonize with the chassis body design, able to installed with the tractor chassis by bolts and nuts, and closed to the vehicle which leads to the calculation of the weight and the suitable volume of the new counter weight to keep the balance of the vehicle to prevent the rollover characteristic when operating in the sugarcane farm. As the result, the weight of the counter weight is 1,000 kg, and the inside of the counter weight contains steel ball for the easiness to fill and refill the weight.

4. STATIC FORCE ANALYSIS

4.1 Static force analysis of loader arm

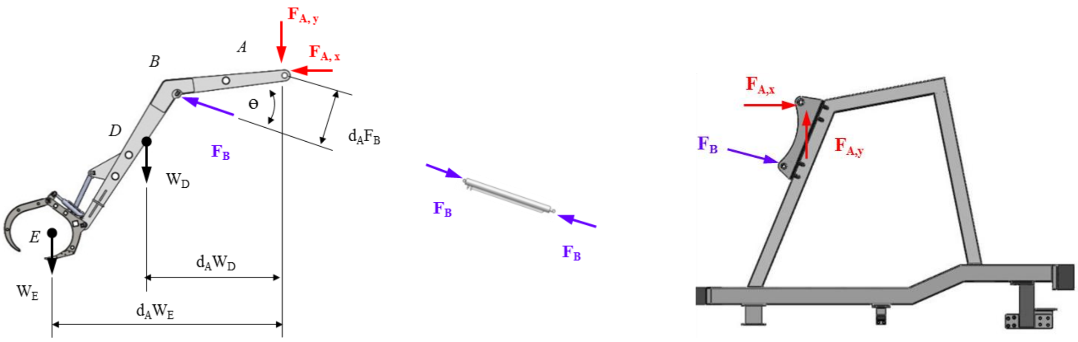
Before study on structural analysis of sugarcane loader structure by FEM, the maximum force that acts to the structure must be determined first, which these load will be defined as severe boundary condition in Finite Element Analysis (FEA) [1, 2]. Considering to the free body diagram as shown in Fig. 2 (a), there are 4 main forces that act to the loader arm: First, the force acting to the loader arm which comes from the summation of weight of loader arm and grabber (W_D) and the magnitude is 675 kg. Second, the force acting to the loader arm which comes from the weight of sugarcane bundle (W_E) and the magnitude is 300 kg. Third, the force acting by the hydraulic cylinder at point B (F_B). And fourth, the reaction force at the joint between the loader arm and the loader structure at point A (F_A). The force at point A and point B can be determined by equilibrium equations as below.

$$F_B = \frac{W_D dW_D + W_E dW_E}{d_A F_B} \tag{1}$$

$$F_{A,x} = -F_B \cos\theta \tag{2}$$

$$F_{A,y} = F_B \sin\theta - W_E - W_D \tag{3}$$

$$F_A = \sqrt{F^2{}_{A,x} + F^2{}_{A,y}} \tag{4}$$



(a) (b)

Fig. 2. Free Body Diagram, (a) loader arm, (b) loader structure.

Where, $d_A W_E$ is perpendicular distance with W_E from the hinge at point A, $d_A W_D$ is perpendicular distance with W_D from the hinge at point A, $d_A F_B$ is perpendicular distance with force acting by the hydraulic cylinders (F_B) from the hinge at point A.

Respecting to the design concept, the maximum vertical lifting distance of the grabber is 5 meters from the ground, it influences the weight of grabber, loader arm and sugarcane bundle acting to the hinge points ($F_{A,x}$ and $F_{A,y}$), and the force from hydraulic cylinder (F_B) will be varied regarding to the angle between hydraulic cylinder and the horizontal plane (θ), from 12.54 degree (at minimum hydraulic cylinder stroke, 0 mm) to 81.98 degree (at maximum hydraulic cylinder stroke, 699 mm). As the result, the maximum angle is 81.98 degree (at maximum hydraulic cylinder stroke, 699 mm) creates maximum reaction force at point A and B and the magnitude of reaction force at point A (F_A) and B (F_B) are 56,373.15 N and 65,828.56 N respectively. Moreover, reaction force at point A (F_A) can be express in x and y axis which the value in x-axis ($F_{A,x}$) and y-axis ($F_{A,y}$) are -9,184.32 N55,619.97 N respectively. Reaction force at point B (F_B) can be express in x and y axis which the value in x-axis ($F_{B,x}$) and y-axis ($F_{B,y}$) are -9,184.32 N and 65,373.15 N respectively.

4.2 Static force analysis of loader structure

Considering to the free body diagram of loader structure as shown in Fig. 2 (b), there are 2 main forces that act to the loader structure: First, the force acting to the hinge point A of the loader arm ($F_{A,x}$ and $F_{A,y}$). Second, the force from the hydraulic cylinder (F_B). From the equilibrium equation as shown in Eqs. (1) - (4), it can define the reaction force at point A and B as well as static force analysis that act to the loader arm, which are the same values but opposite directions.

5. STRENGTH ANALYSIS BY FINITE ELEMENT METHOD

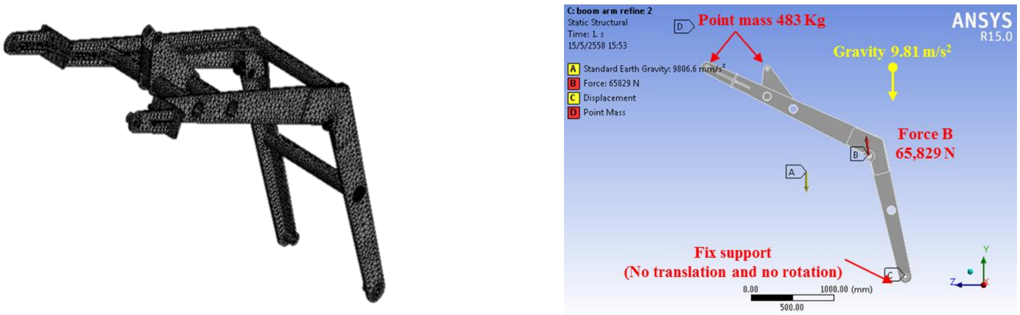
From the analysis of the static force acting to Sugarcane loader structure to find the severe condition, at the highest lifting point of the grabber (or at maximum hydraulic cylinder stroke, 699 mm), it will create maximum static force acting to loader structure and loader arm. Therefore, the next step is to analyze the strength of the designed parts by follow this severe condition. In this research, the material of the sugarcane loader is ductility material. Therefore, designed part is analyzed following the Von Mises Yield Criterion because it is more reliable in results. The Von Mises stress value should smaller than the yield strength of the material divided by safety factor, which in this research, the safety factor is 3 [3, 4]. If the result does not follow the Von Mises Yield Criterion, the designed parts will fail and need to be improved strength until they pass this criterion [5-9]. The Von Mises Criterion as shown in Eq. (5).

$$\sigma_{von} \le \frac{S_y}{S.F.} \tag{5}$$

Where, σ_{von} is Von Mises stress (MPa), *Sy* is Yield strength of material (MPa), *S.F.* is safety factor.

5.1 Strength analysis of loader arm

The loader arm 3D model is created by SolidWorks software. This characteristic is double grabber arms which consist with one single part. The element type of loader arm is Tetrahedral. The minimum element size is 2.5 mm which is selected by mesh sensitivity analysis [10, 11]. Number of total nodes is 505,360 nodes and number of total element is 303,788 elements as shown in Fig. 3 (a).



(a) (b)

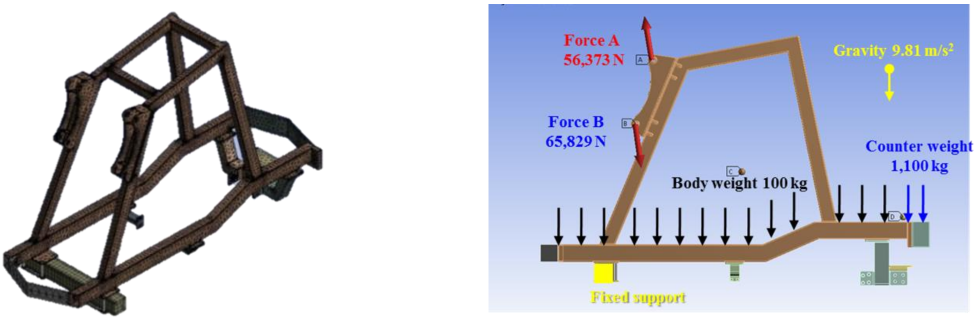
Fig. 3. FE model of loader arm, (a) meshing, (b) boundary condition.

There are 3 loads and 1 constraint are applied to the FE model of loader arm: 1) Summation of weight of the grabber and sugarcane bundle, 483 kg is applied to the pin-hole that attached grabber with loader arm and attached loader arm with hydraulic cylinder of grabber. The weight is specified by Point mass feature in ANSYS software. 2) Hydraulic cylinder force at point B is 65,829N which is applied at the pin-hole that attached loader arm with hydraulic cylinder by force feature in ANSYS. 3) Gravitational acceleration is 9.81 m/s^2, is applied in vertical direction to the whole loader arm model. 4) Specify the constraint of loader arm model by fixed support around the pin-hole of assemble point of the loader arm and the loader structure. The loader arm will not possible to have any movement in X, Y, and Z axis and it is not able to rotate around X, Y, and Z axis. The loads and boundary condition are shown in Fig 3 (b).

The material of the loader arm is ss400 Carbon steel tube. The behavior of ss400 steel is linear elastic. Density is 7,850 kg/m^3. Young modulus is 205 GPa. Poisson's ratio is 0.29. Yield strength is 250 MPa.

5.2 Strength analysis of loader structure

The loader structure 3D model is created by SolidWorks software. It contains 11 main parts: 1) Front cross member, 2) Rear cross member, 3) Frame, 4) Right engine-frame support, 5) Left engine-frame support, 6) Right transmission-frame support, 7) Left transmission-frame support, 8) Counter weight support, 9) Left Plate, 10) Right Plate, 11) Rigid beam. The rigid beam is acting as the core of rear driveshaft as to support the loader structure. Moreover, the actual producing parts will be assembled by welding. Therefore, the welding lines are created by using fillet feature in SolidWorks software as to make the designed model more complete and to avoid the stress singularity. The defined radiuses of fillets are the same value as radius of welding lines in 2D drawing. The element type of loader structure is Tetrahedral. The minimum element size is 2.5 mm which is selected by mesh sensitivity analysis. Number of total nodes is 730,704 nodes and number of total element is 463,374 elements as shown in Fig. 4 (a).



(a) (b)

Fig. 4. FE model of loader structure, (a) meshing, (b) boundary condition.

As there are 11 parts of the loader structure and each part can be detach, therefore, the contact condition is applied by Frictional-contacts feature in ANSYS software which two faces of each component are free to separate in their normal direction as well as sliding in their tangential direction and friction coefficient between each parts, μ is 0.74, which is friction coefficient of steel to steel. Moreover, as each component of loader structure is designed to assemble by bolts and nuts, therefore, Beam-contacts feature in ANSYS software is applied to simulate as fasteners with realistic bolt diameter.

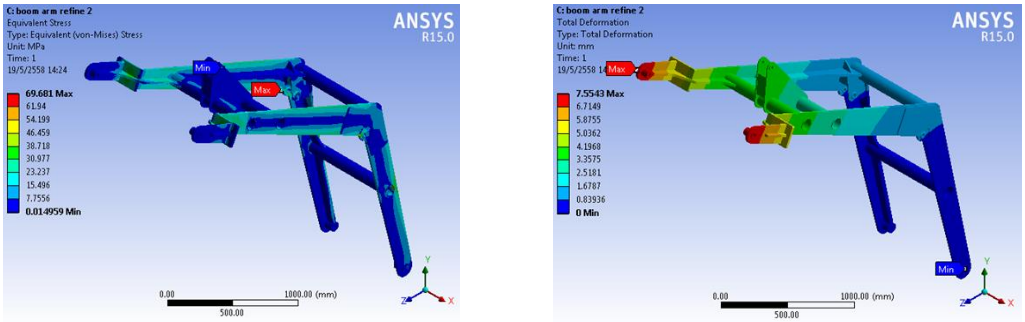
There are 4 main loads and 1 constraint acting to the loader structure model: 1) Longitudinal load from weight of body parts, 100kg is applied on the frame by Point mass feature in ANSYS. 2) Load from weight of the counter weight, 1,100 kg is applied at the attached point of the counter weight to the frame by Point mass feature in ANSYS. 3) Force at point A is 55,373 N, and point B is 65,829 N. The forces are applied by Force feature in ANSYS. 4) Gravitational acceleration is 9.81 m/s^2 which acting to the whole loader structure. 5) The loader structure model is attached by fixed supports around the side area of the rigid beam. In other words, there is no movement and no rotation in 3 axles (X, Y, and Z) which is assumed as the sugarcane loader structure is placed on the rigid beam virtually and the boundary conditions are shown in Fig. 4 (b). Moreover, the material of loader structure is ss400 steel which is the same as loader arm.

6. Result and Discussion

6.1 Strength analysis result of loader arm

From Fig. 5 (a) the maximum Von Mises stress of the loader arm around the pin hole of hydraulic cylinder is 69.68 MPa. Considering to Von Mises stress criterion, the specific chosen designed safety factor is equal or more than 3 (*S.F.* ≥ 3), the maximum Von Mises stress is 69.68 MPa (σ_{von} = 69.68 MPa) which is smaller than the yield strength of steel ss400 (S_y = 250 MPa). Therefore, the calculated value of safety factor is 3.59 (*S.F.* = 3.59), which is more than the original specified safety factor, which mean that the designed parts are safety to use.

In addition, the maximum displacement is 7.55 mm which occurred at the near end of the loader arm closed to the pin-hole that assembled the grabber as shown in Fig. 5 (b) and it is less displacement value when compared to the minimum thickness of the steel plate that used for the loader arm (Thickness is 19 mm). Therefore, the design of the loader arm is safe to apply loading condition.



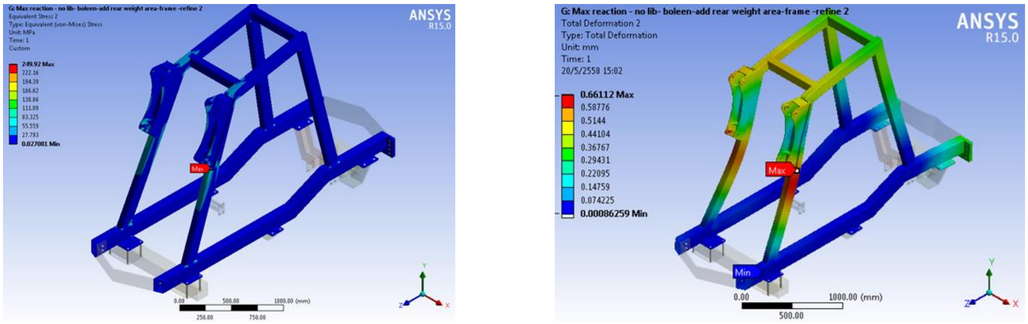
(a) (b)

Fig. 5. Strength analysis result of loader arm, (a) Von Mises stress, (b) deformation**.**

6.2 Strength analysis result of loader structure

From Fig. 6 (a) the maximum Von Mises stress of the loader structure is 249.92 MPa acting at hydraulic cylinder mounting lug of the frame. When considering the failure criterion theory, as respecting to the specified designed safety factor is equal or more than 3 ($S.F. \geq 3$), the maximum Von Mises stress is 249.92 MPa (σ_{von} = 249.92 MPa), which is very closed to the yield strength of steel ss400 (S_y = 250 MPa), it represented the safety factor at 1.00 ($S.F.$ = 1.00) which is less than the specified designed safety factor ($S.F.$ = 3). Therefore, the designed loader structure is not safe enough to use and it needs to be improved strength.

Future more, the maximum displacement of loader structure has is 0.661 mm which occurred around the area that taking the force from the hydraulic cylinder as well as the maximum stress as shown in Fig. 6 (b).



(a) (b)

Fig. 6. Strength analysis result of frame of loader structure, (a) Von Mises stress, (b) deformation.

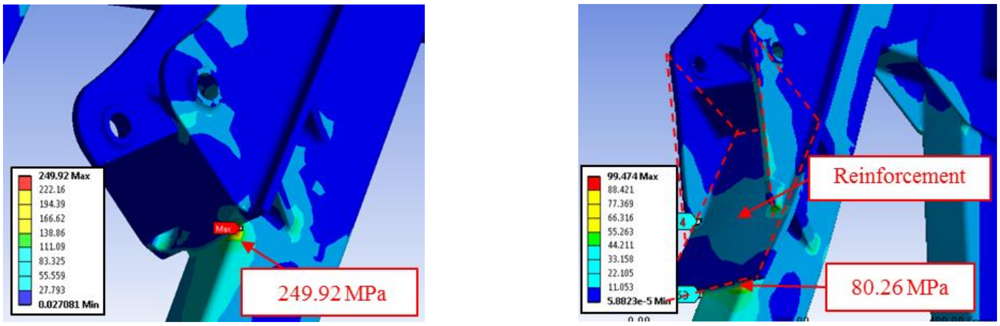
6.3 Strength improvement of loader structure

To reduce the stress on critical point of the loader structure, the idea of strength improvement is increasing the moment of inertia of section by changing the shape of loader structure at critical point [12]. Regarding to the produced stress at critical point of the loader structure is bending stress which is effect from bending moment. As bending stress in Eq. (6), if moment of inertia of section is increased, bending stress will be decreased.

$$\sigma_{bending} = \frac{My}{I} \tag{6}$$

Where, $\sigma_{bending}$ is Bending stress (MPa), M is Bending moment (N.mm), y is Height of section, I is Moment of inertia (mm^4)

To reduce stress on critical point, the loader arm supporting of the frame is modified by adding reinforcement, as shown in Fig. 7. The Von Mises stress on critical point is decreased from 249.92 MPa to 80.26 MPa. Moreover, the safety factor is increased from 1.00 to 3.11 which the value is more than the specific safety factor, 3. Therefore, the designed loader structure is safe to withstand the load.



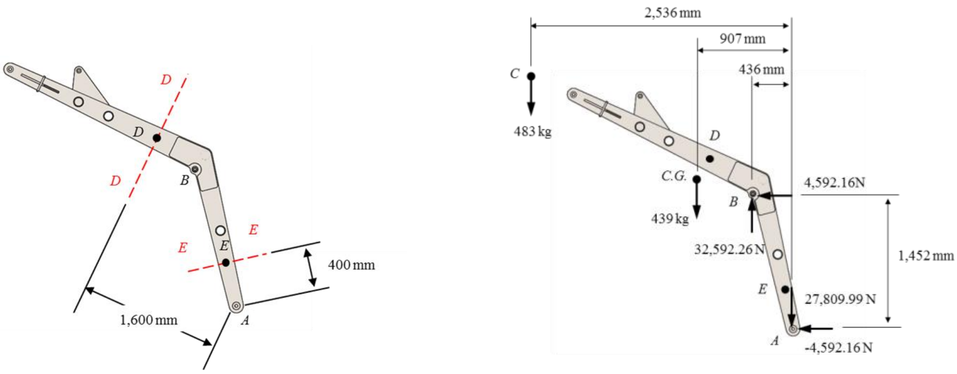
(a) (b)

Fig. 7. Stress result on critical point of loader structure, (a) before improvement, (b) after improvement.

6.4 Verification of stress analysis using classical theory

As this work does not study the stress measurement on the prototype of sugarcane loader structure by strain gauge due to limitation of time and instrument, therefore, the produced stress on Finite Element Analysis by computational simulation is verified by classical theory; Engineering Mechanic Static, Solid Mechanic, and Machine Design, with the same cross-section in designed part [13].

The sugarcane loader structure consists of two main components; loader arm and loader structure. As the limitation of classical stress analysis on loader structure; complex shapes and statically indeterminate structure (Static equilibrium equations are insufficient for determining the internal loads and reaction loads on the structure), therefore, in this section is focused on the verification of stress analysis using classical theory on the loader arm only. Regarding to the Von Mises failure theory of ductile material that defined as design criteria, therefore the produced Von Mises stress at point D on cross-section D-D and at point E on cross-section E-E of the loader arm, which is randomly selected, are used to compare between FEA approach and classical theory. The cross-section D-D is located at the middle of the loader arm which is taken at 1,600 mm from point A and the cross-section E-E is located between point A and point B which is taken at 400 mm from point B as shown in Fig. 8 (a). In order to make the stress calculation by classical theory easily, the external force and weight acting on the loader arm are divided by 2 because the loader arm is symmetry geometry and free body diagram is shown in Fig. 8 (b).



(a) (b)

Fig. 8. (a) Section D-D and E-E on loader arm, (b) Free body diagram of loader arm.

6.4.1 Stress analysis of loader arm on cross-section D-D using classical theory

The internal loadings at point D on cross-section D-D are found out by using free body diagram of the section CD of loader arm shows in Fig. 9. There are three internal loadings: Normal force (N_D), acting normal to loader arm at the cut section, Shear force (V_D), acting tangent to the section and Bending moment (M_D). The magnitudes of those internal loading are determined by Equation of Equilibrium as below,

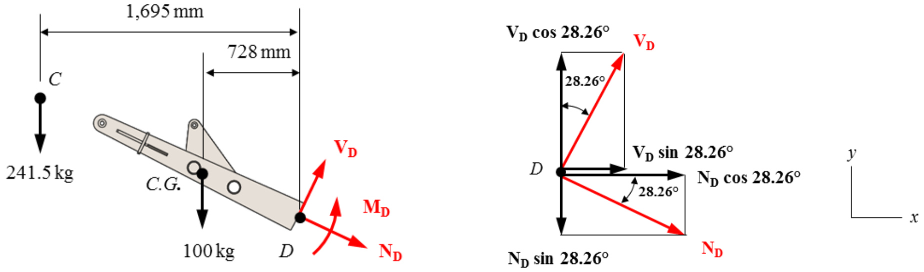


Fig. 9. Free body diagram of loader arm on cross-section D-D.

Equations of Equilibrium,

$$+\downarrow \sum F_y = 0;\ N_D = 1.83V_D - 6994N \tag{7}$$

$$\overset{+}{\rightarrow} \sum F_x = 0;\ V_D = -1.83N_D \tag{8}$$

Substituted Eq. (8) in Eq. (7),

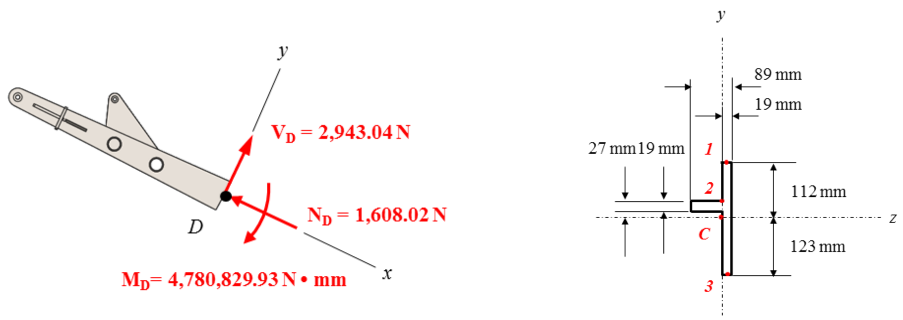
$$N_D = -1608.22\text{N}$$

$$V_D = 2943.04\text{N}$$

$$\sum M_D = 0;\ M_D = -4780829.93\text{N}\cdot\text{mm} \tag{9}$$

Summary of magnitude and direction of internal loadings at point D (cross-section D-D) is shown in Fig. 10 (a)

As geometric properties of section D-D in Fig. 10 (b),
Moment of inertia of the cross-section D-D about z –axis, I_z = 21239729 mm^4
Area of the cross-section D-D, A = (89mm - 19mm)(19mm) + (112mm + 123mm)(19mm) = 5795 mm^2



(a) (b)

Fig. 10. Cross-section D-D of loader arm, (a) forces acting on section, (b) Geometric section.

Stress due to axial force and bending moment can be determined by Eq. (10),

$$\sigma_x = \sigma_{axial} + \sigma_{bending} = \frac{N}{A} + \frac{M_z y}{I_z} \tag{10}$$

Shear stress can be determined by Eq. (11),

$$\tau_{xy} = \frac{V}{A} \tag{11}$$

Von Mises stress can be determined by Eq. (12),

$$\sigma_{von\ mises} = \sqrt{\sigma^2{}_x - \sigma_x\sigma_y + \sigma^2{}_y + 3\tau^2{}_{xy}} \tag{12}$$

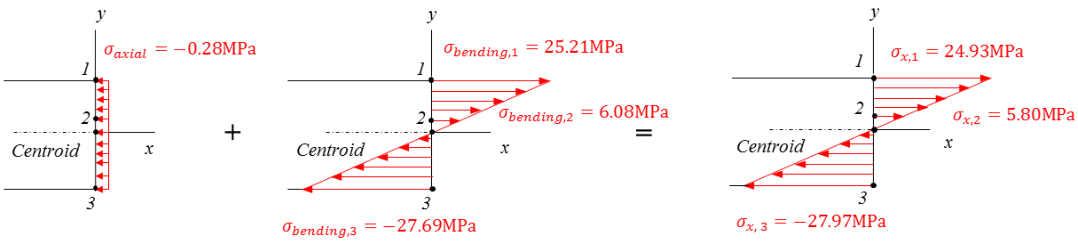
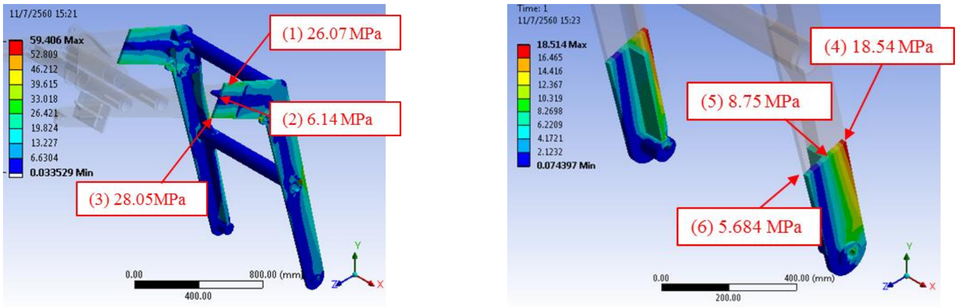


Fig. 11. Stresses due to axial force and bending moment on section D-D of loader arm.

As following Eqs. (10) - (12), the magnitudes of Von Mises stress at point number 1 (at upper surface of cross-section D-D), point number 2 (at middle of cross-section D-D) and point number 3 (at below of cross-section D-D) by classical theory are 24.95 MPa, 5.87 MPa and 27.98 MPa respectively. Moreover, as the Von Mises stresses of loader arm have been found out by FEA approach, the result shows that the magnitudes of stress at the point number 1, 2, and 3 are 26.07 MPa, 6.14 MPa and 28.05 MPa respectively, which are shown in Fig. 12 (a). It clearly indicates that the stresses between FEA and classical theory are very close.



(a) (b)

Fig. 12. Stress results on loader arm by FEA (a) section D-D (b) section E-E.

6.4.2 Stress analysis of loader arm on cross-section E-E using classical theory

The internal loadings at point E on cross-section E-E are found by using free body diagram of the section CE of loader arm as shown in Fig. 13. There are three internal loadings: Normal force (N_E), acting normal to loader arm at the cut section, Shear force (V_E), acting tangent to the section and Bending moment (M_E). The magnitudes of those internal loading are determined by Equation of Equilibrium as below,

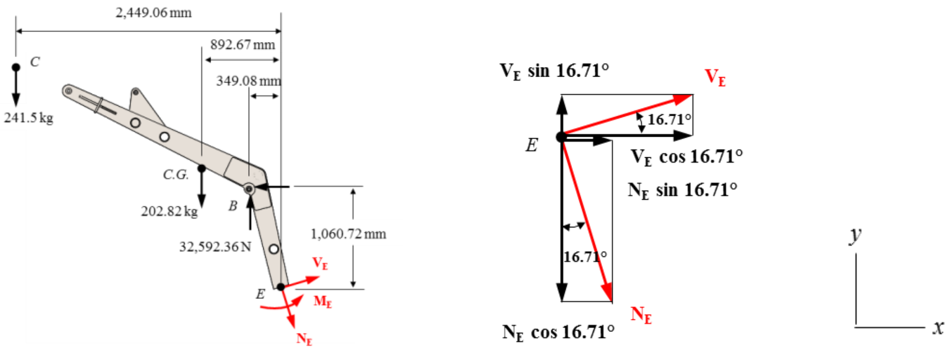


Fig. 13. Free body diagram of loader arm on cross-section E-E.

Equations of Equilibrium,

$$+\downarrow \sum F_y = 0;\ N_E = 0.3V_E + 28233.58N \tag{13}$$

$$\overset{+}{\rightarrow} \sum F_x = 0;\ N_E = -3.33V_E + 15971.19N \tag{14}$$

Eq. (13) equals Eq. (14),

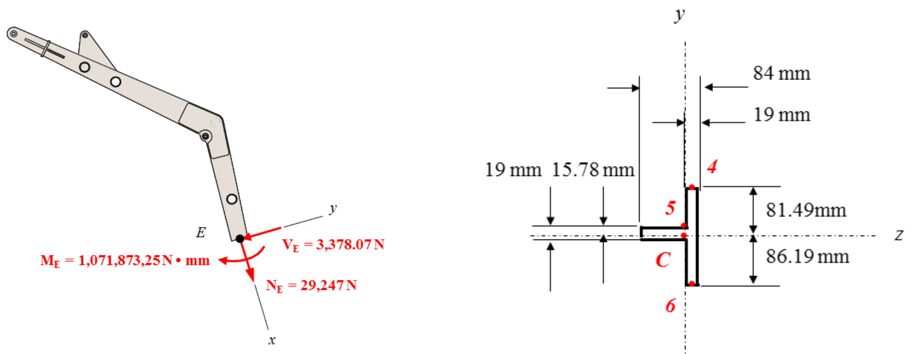
$V_E = -3378.07\text{N}$

$N_E = 29247\text{N}$

$$\sum M_E = 0;\ M_E = -1071873.25\text{N}\cdot\text{mm} \tag{15}$$

Summary of magnitude and direction of internal loadings at point E (cross-section E-E) is shown in Fig. 14 (a)

As geometric properties of section E-E in Fig. 14 (b),
Moment of inertia of the cross-section E-E about z –axis, I_z = 7565551 mm^4
Area of the cross-section E-E, A = (84 mm - 19 mm)(19 mm) + (81.49 mm + 86.19 mm)(19 mm) = 4420.92 mm^2



(a) (b)

Fig. 14. Cross-section E-E of loader arm, (a) forces acting on section, (b) Geometric section.

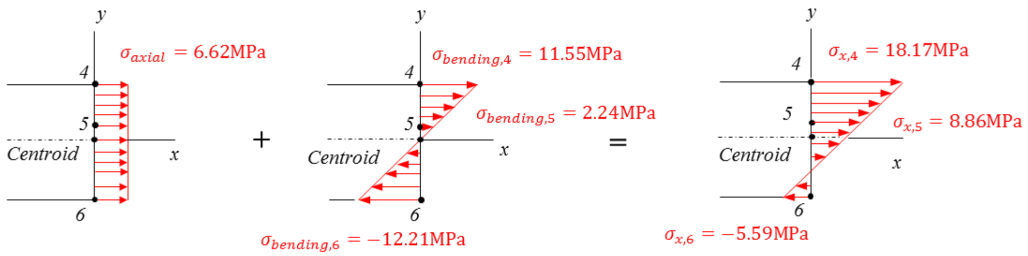


Fig. 15. Stresses due to axial force and bending moment on section E-E of loader arm.

As following Eqs (10) - (12), The magnitudes of Von Mises stress at point number 4 (at upper surface of cross-section E-E), point number 5 (at middle of cross-section E-E), and point number 6 (at below of cross-section E-E) by classical theory are 18.22 MPa, 8.96 MPa, and 5.74 MPa respectively. Moreover, as the Von Mises stresses of loader arm have found out by FEA approach, the results show that the magnitudes of stress at point number 4, 5 and 6 (on cross-section E-E) are 18.54 MPa, 8.75 MPa, and 5.68 MPa respectively, which are shown in Fig. 12 (b). It clearly indicates that the stresses between FEA and classical theory are very close.

From Fig. 16, the comparison of the Von mises stresses at point number 1, 2, 3, 4, 5, and 6 (on cross-section D-D and cross-section E-E) of the loader arm between classical theory and FEA, there are very similar trend of the stresses between the classical theory and FEA. As result, the Von Mises stresses of the loader arm from classical theory at point number 1, 2, 3, 4, 5, and 6 (on cross-section D-D and E-E) are 24.95 MPa, 5.87 MPa, 27.98 MPa, 18.22 MPa, 8.96 MPa, and 5.74 MPa respectively. The Von Mises stresses of the loader arm from FEA at point number 1, 2, 3, 4, 5, and 6 are 26.07 MPa, 6.14 MPa, 28.05 MPa, 18.54 MPa, 8.75 MPa, and 5.68 MPa respectively, which they have found out in strength analysis results of loader arm by Finite Element Method, which it clearly indicates that the stresses from classical theory and FEA are very close signicicantly.

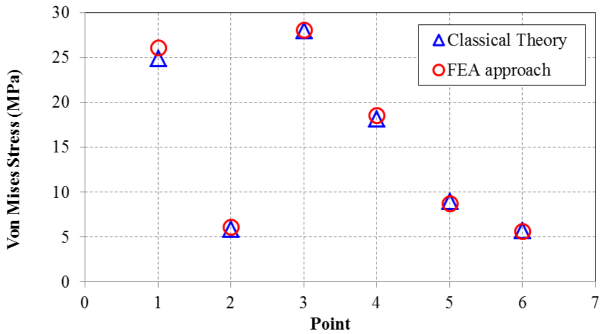


Fig. 16. Von Mises stresses on section D-D and E-E of loader arm between classical theory and FEA.

7. Conclusions

According to Static Force Analysis, maximum hydraulic cylinder force acting on sugarcane loader structure occurs when the grab operated at maximum hydraulic cylinder stroke condition. Therefore maximum stroke of hydraulic cylinder condition) can be used as the severe boundary condition for static FEA of sugarcane loader structure.

Actually, Thai entrepreneur has not designed the sugarcane loader structure base on engineering knowledge. The sugarcane loader structure always fails in field. After using FEM in design process, static analysis of sugarcane loader structure by using FEA approach can predict stress developed on parts before the prototype fabrication. From the results, the maximum stress occurs around the hydraulic cylinder mounting lug of frame due to bending effect and stress concentration. Therefore designer should concern more on strength of this area. Moreover, the sugarcane loader structure is found to be safe under maximum hydraulic force condition by performing static FEA. The final safety factors of loader arm and loader structure are 3.59 and 3.11 respectively.

As this work does not study the stress measurement on the prototype of sugarcane loader structure by strain gauge because of time and instrument limitation, thus, the produced stress on FEA by computational simulation is verified by classical theory with the same section of designed parts. However, the limitation of stress analysis on loader structure by classical theory; complex shapes and in state of statically indeterminate structure (Static equilibrium equations are insufficient for determining the internal loads and the reaction loads on the structure) thus, the stress verification by classical theory is studied on the loader arm only. From the results, stress on section D-D and E-E of loader arm from FEA and classical theory are very close and the percentage of deviations at point number 1, 2, 3, 4, 5 and 6 of loader arm are 4.53%, 4.60%, 0.25%, 1.76, 2.34% and 1.05% respectively, which the deviations are really less. Therefore the produced stresses by FEA approach can be verified by applying the classical theory.

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