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Research Article

Analysis and Topological Optimization of Motorcycle Swing-Arm

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Abstract

This article is based on optimization of a motorcycle swing arm. The modification process is based on material, topological modification and validation using finite element analysis. The results obtained from modified analysis are compared with the evaluation of the original component. The goal of the experiment is to reduce the mass of the component without compromising the other relevant factors. For analysis and study, a well reputed general class 150 cc motorcycle's swing arm was selected.

Keywords: *Swing-arm, topological modification and validation.*

1. Introduction

[1]IC engine powered vehicles have a long history and are still dominant in its segment. These engines use fossil fuels mainly petroleum oils and gases as fuels having higher calorific values. Gasoline, diesel oils and natural gases are widely used on regular basis. These are being used continuously since a long time ago and continue to be explored. Although the development of new sources is in progress, there surely is a need to retard the demand of non-renewable energy sources. The process of burning gasoline to power cars and trucks contributes to air pollution by releasing a variety of emissions into the atmosphere. India was the sixth largest motor vehicle/car manufacturer in the world in 2013(wikipedia). India is the second largest motorcycle (6.54 m produced in 2007-08) and the fourth largest commercial vehicle manufacturer in the world. This shows that motorcycles are major contributors to the overall vehicles. They share a large part of total daily fuel consumption of our country. An average human weighs about 65-75 kg. The combined weight of motorcycle and rider would be near about 200-220 kg. Hence we can say that about 70 % of fuel is consumed by the motorcycle itself. Obviously this cannot be eliminated nor can be drastically reduced but a slight reduction in one two-wheeler will cause a significant impact upon overall fuel consumed by the same model all over the country.

2. Nomenclature

L_s Static load per side beam
M_t Mass of motorcycle
m_p Average mass of person
L_{vs} Vertical Load on side beam
L_{vh} Horizontal load on side beam
F_{iH} Horizontal Load On inner horizontal side
F_{oH} Horizontal Load On outer horizontal side
θs Spring damper inclination
S_{yt} Yield strength Of material
ab Maximum Braking deceleration
F_{iV} Vertical Load on Inner Vertical Side
F_{oV} Vertical Load on outer Vertical Side

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3. Swing Arm

The motorcycle Swingarm is a key component of the rear suspension of a motorcycle. It connects the rear wheel of the motorcycle to the main chassis and it regulates the rear wheel-road interactions via the spring and shock absorber. Two basic designs exist, namely the single-sided and double-sided swing arms. The vertical stiffness can affect the motorcycle setup and produce unpredictable behavior if not rigid enough. The aim is to maximize the vertical stiffness and ensure it is considerably higher than the rear suspension spring stiffness (B Smith, *et al*, 2015). The lateral and torsional stiffness affect the motorcycle response during cornering and the motorcycle weave mode. The weave mode is the side to side movement of the rear of the motorcycle caused by the roll and yaw motion of the motorcycle. In general, it is desirable to maximize the Swingarm lateral and torsional stiffness to reduce this instability (B Smith, *et al*, 2015).

CAD modeling of the swing arm was done using CATIA V5 software. A reverse engineering approach was used to model the same. The current swing arm is being manufactured by welding different structures together. The original weight was 3.329 kg whereas the cad model weight showed 3.339 kg. Hence the accuracy of the model is 97.1%. This may not be totally justified as there is coating of some protective material such as paint.

4. Material modifications

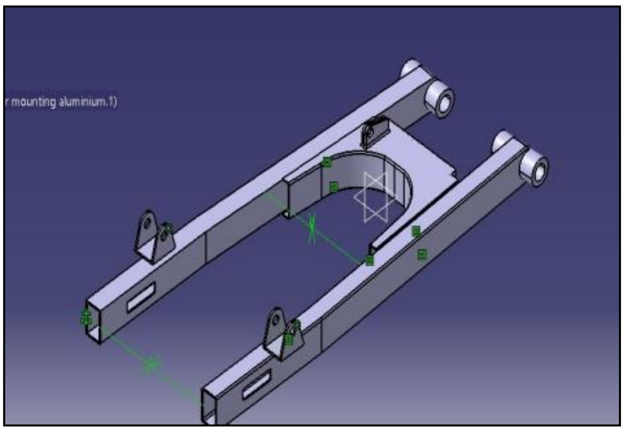
Original material used in the swing arm is AISI 1010. The material used in the modified swing arm is Al 7075. It is used to increase the overall strength of the swing arm keeping an eye on the weight of material. Aluminium is used instead of steel because it can withstand heavy loads in the form of forces compared to the steel swing arm with the advantage being lower density of Aluminium compared to the steel component. Following is the comparison of the previous material and the new material (J. Janardhan, *et al*, 2014). As, Al 7075 has higher strength, it has a scope for weight reduction through geometry modification.

Table 1 Material Comparison (J. Janardhan, *et al*, 2014)

Sr. No	Property	AISI 1010	Al 7075
1	Tensile strength (MPa)	365	510
2	Yield Strength (MPa)	305	434
3	Elastic modulus (GPa)	210	72
4	Poisson's ratio	0.3	0.33
5	BHN	105	60

5. Geometry modifications

Based upon the stress distribution of original swing arm, the geometry of the swing arm is modified. The original cross section is hollow rectangular. For modified geometry, and I section cross section is used. Also, X section ribs are provided in the hollow gap to increase the stiffness. The central joining member is not modified since it is only meant for attachment of the two arms. Following figure shows the modified swing arm CAD model. The mass of one side beam was found to be 0.7035 Kg and that of middle part as 0.197 kg. Thus overall weight comes to be as 1.604 kg. Comparatively, the original swing arm weight was nearly 3 kg.



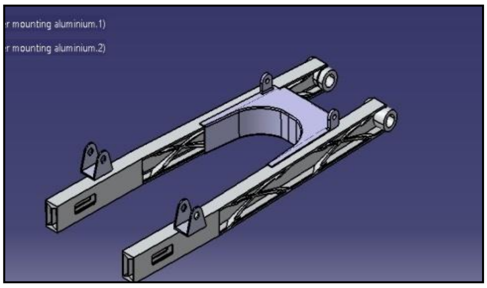


Fig.1 CAD models

6. Analysis

FEA analysis is performed on both the original and modified components. As the swing arm is symmetric, for purpose of analysis only one beam of either side is used (Madhu K S, *et al*, 2014). Rear damper mounting plate is welded on this beam. This is necessary to reduce computational time, memory and energy and also to increase the accuracy of results. ANSYS 15 software is used for analysis and simulation of the components. The meshing used was auto mesh with 4 mm default size. The number of nodes generated was 142884 and number elements generated were 77844 for modified side beam. Following image shows the meshed model of the side beam.

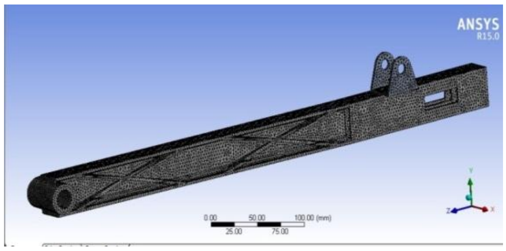


Fig.2 Meshed model of modified side beam

6.1 Weight and Acceleration

The swing arm has cylindrical frictionless supports attached to the chassis and other end has the bearings in which the rear wheel axle is rotating. The spring-dampers are mounted on the welded plate. During static running condition, the dampers exert forces due to the dead weight of bike and people on the swing arm, which acts on the rear side of the motorcycle (B Smith, *et al*, 2015). Also during maximum acceleration, the chain exerts torque on the sprocket. This load acts as pressure on the swing arm on rear lateral faces where the wheel hub is mounted. Considering these two conditions, one critical condition could be the simultaneous application of these two loads. This condition needs to be analyzed.

Loads and boundary conditions- The weight of the motorcycle is 143 kg. Considering average weight of person as 75 kg, total dead weight is 293 kg. In most two-wheelers, the distribution of weight on rear axle is 58% to 65%. For the model selected, the weight distribution is taken to be 60 % on rear axle. Also 30% of weight is reduced due to tires and wheels and other unsprung masses.

Thus net load on swing arm can be calculated as,

$Ls= [m_s + 2m_p] \times 0.6(1)$ = 193.8 kg.

This 193.8 kg which will be distributed equally on the two side beams in case when the motorcycle is running straight. The load will be acting at an angle of about 53^o at which the damper is mounted.

Thus, the loads are separated into vertical and horizontal components.

Vertical load $L_{vs}=L_s Sin\ \theta_s$ & horizontal load $L_{vh}=L_s Cos\ \theta_s$. *i.e.* L_{vs} = 1456.4 N and L_{vh} = 1222.13 N.
The maximum acceleration of the motorcycle is found to be a = 5m/s^2.

Also total mass m_t = 293 kg. Hence longitudinal force acting on the swing arm can be found as

$F_L = m_T \times a(1)$,

This is calculated as 1465 N.

Also, the cross sectional area on which acceleration force is acting is found to be 22 mm^2. Thus pressure value becomes 12.22 MPa for modified beam. These loads are applied to the side beam as shown in figure 3.

For this condition, the equivalent stress generated in the original side beam was 186.32 MPa. The stress generated in the modified side beam is 151.66 MPa as shown in figure 4. This value is well below the yield strength. The factor of safety can be calculated as $N_f = S_{ut}/\sigma_{max}$ = 2.789.

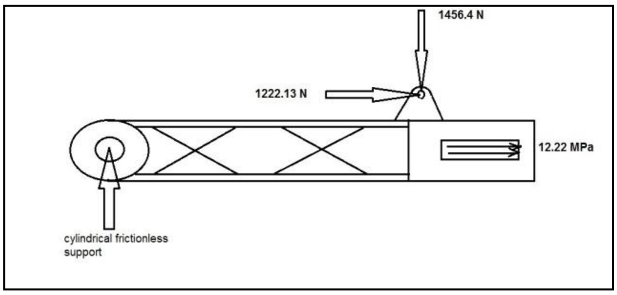


Fig.3 Loads and constraints for weight and acceleration

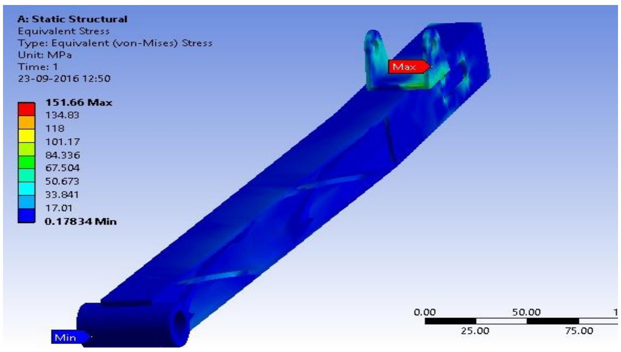


Fig.4 Equivalent stress during acceleration

6.2 Weight and Braking Condition

This condition is similar to the one mentioned above the difference being that the pressure due to braking will be in opposite direction. The minimum braking time was evaluated experimentally and maximum deceleration was found. For rear braking, the maximum deceleration was found when braking from 20 kmph to 0 in 1.3 seconds. From this value, the maximum deceleration is -4.273 m/s^2. Considering the inertia of the bike and this acceleration (B Smith, *et al* 2015), the longitudinal force on swing arm is

$F_L = m_t \times a_b$ *i.e.* 293 × (-4.273) = -1252 N (negative sign indicates force acting in backward direction).

Again considering area of cross-section, longitudinal pressure on Swingarm is 10.44 MPa. This pressure is applied with boundary conditions as in first case.

The equivalent stress is found to be 102.8 MPa which is less than yield strength of the material, with a factor of safety of 4.2. In case of the original swing-arm, the value of equivalent stress was 130.05 MPa as shown in figure 5. Hence factor of safety in braking is 4.11.

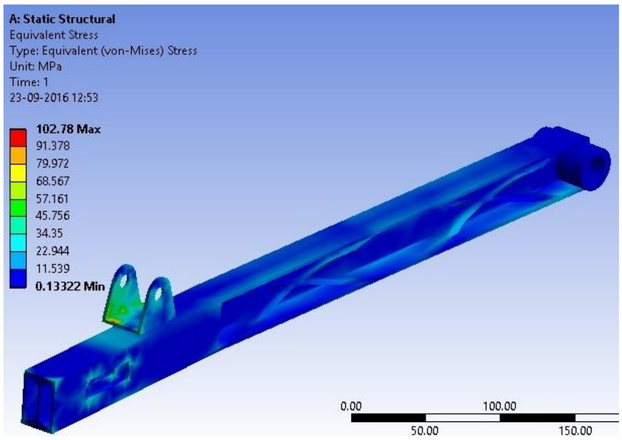


Fig.5 Equivalent stresses during braking

6.3 Cornering Condition

Cornering is one of the important criteria in design on motorcycle components. During cornering, different components are subjected to variation in loads in magnitude as well as direction. In case of swing arm, high lateral forces act in unbalanced state. The magnitude of variation depends upon the angle of inclination and the vehicle speed (B Smith, *et al,* 2015).

Loads and boundary conditions- It is assumed that 20% more load are transferred to the inner side during cornering. Thus, the inner side beam will have 70% of the total weight and remaining 30% on the outer side beam. If we consider a maximum cornering angle of 40^o, and divide the forces into vertical and horizontal components, there will be torsional and lateral imbalance on the middle part (B Smith, *et al*, 2015).

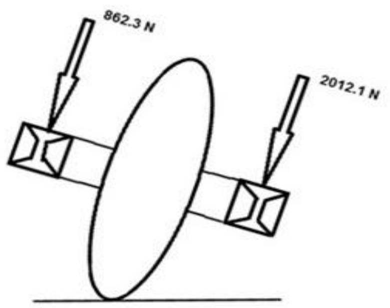


Fig.6 Cornering condition

70% of weight = F_{max}= 0.7×293×9.81 = 2012.1 N and remaining 30% = F_{min} = 862.3 N.

Horizontal components (acting as lateral imbalance):

$F_{iH} = F_{max}\,Cos\theta$= 2012.1 Cos 40 = 1541.35 N and

$F_{oH} = F_{min}\,Cos\theta$ = 862.1 Cos 40 = 660.4 N.

Also, vertical components (acting as torsional imbalance):

$F_{iV} = F_{max}\,Sin\theta$ = 2012.1 Sin 40 = 1293.35 N and

$F_{ov} = F_{min}\,Sin\theta$ = 862.1 Sin 40 = 554.15 N.

Thus there are imbalanced forces acting during cornering.

For analysis, a cylindrical frictionless support is applied and forces are applied in components. The maximum values i.e. the inner side swing arm and the middle part is analyzed. The inner side Swingarm will experience more force than outer one. The imbalance will be acting on the middle part.

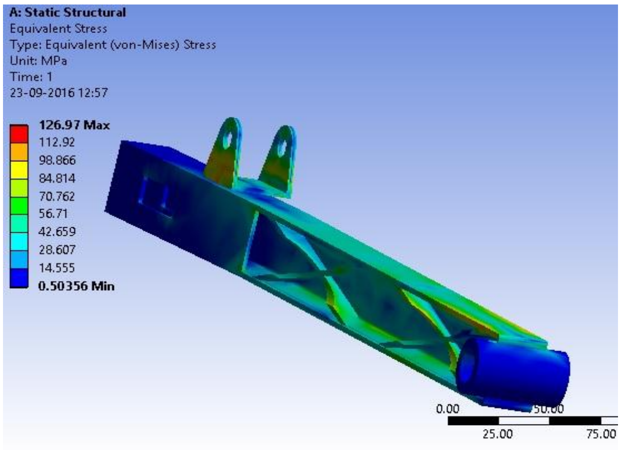


Fig.7 Equivalent stresses during cornering

The equivalent stress on the inner side beam was found to be 126.97 MPa while the maximum principle stress was 134.9 MPa. For the original side beam, the equivalent stress was 152.18 MPa. The factor of safety was found to be 3.13. The stress distribution in the modified side beam was found to be more uniform.

As mentioned earlier, the function of middle part is to only hold the two swing arms. For this purpose, it will have to sustain the lateral unbalancing forces as well as the moment due to torsional unbalancing forces.

The lateral unbalancing equivalent force is calculated as 477.7 N and moment is found to be 55.762 Nm.

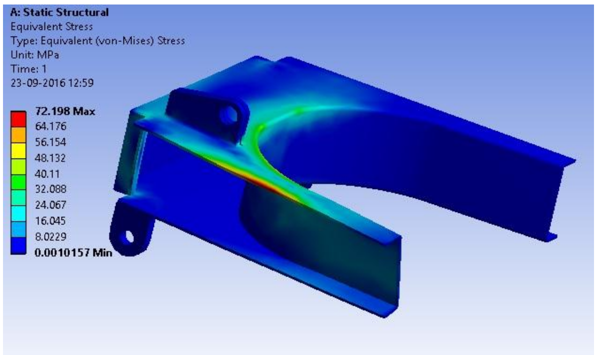


Fig.8 Stresses on mid part in lateral condition

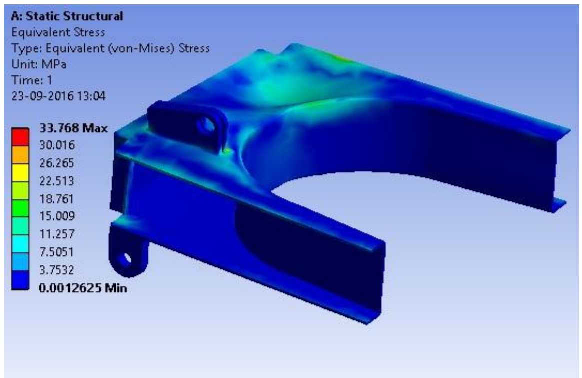


Fig.9 Stresses on mid part in torsional condition

The maximum equivalent stresses were found to be 72.2 MPa and 33.768 MPa in lateral and torsional conditions respectively, using Al 7075. The maximum deformation was 0.219 mm and 0.0157 mm.

Thus the modified swing arm is quite safe in static running conditions provided that the assembly joints and welds have minimum or no defects.

6.4 Fatigue Life Estimation

The process of fatigue failure can be divided into different stages, which, from the stand point of metallurgical processes, can be divided into five stages (Sanup Kumar, 2012):

- Cyclic plastic deformation prior to fatigue crack initiation
- Initiation of one or more micro cracks
- Propagation or coalescence of micro cracks to form one or more micro cracks
- Propagation of one or more macro cracks
- Final failure

S – N Curves- The fatigue properties of any material can be evaluated based on three types of approach as listed below (V. B. Bhandari, 2010).

- Stress-life (*S-N*)
- Strain-life (ε-*N*)

General applicability of the stress-life method is restricted to circumstances where continuum, "no cracks" assumptions can be applied. The advantages of this method are simplicity and ease of application, and it offers some initial perspective on a given situation. It is best applied in or near the elastic range, addressing constant amplitude loading situations in what has been called the long-life regime (Sanup Kumar, 2012).

Most two-wheeler parts are designed for low cycle fatigue. Low-cycle fatigue approach is used where relatively large loads are carried with low fluctuations. For low cycle fatigue theory, strain-life approach needs to be used for which strain life parameters are required. The strain life parameters Al 7075 are tabulated below

Table 3 Strain life parameters Al 7075 (steelforge)

Parameter	Value
Strength coefficient	328.5 MPa
Strength exponent	-0.0739
Ductility coefficient	0.0849
Ductility exponent	-0.42
Cyclic strength coefficient	522.1 MPa
Cyclic strain hardening coefficient	0.2

Fully reversed horizontal and vertical components of weights were applied to the side arm. Cylindrical frictionless support was applied to the other end. Based upon this the safety factor variation and life of the side beam was evaluated. Following are the results obtained

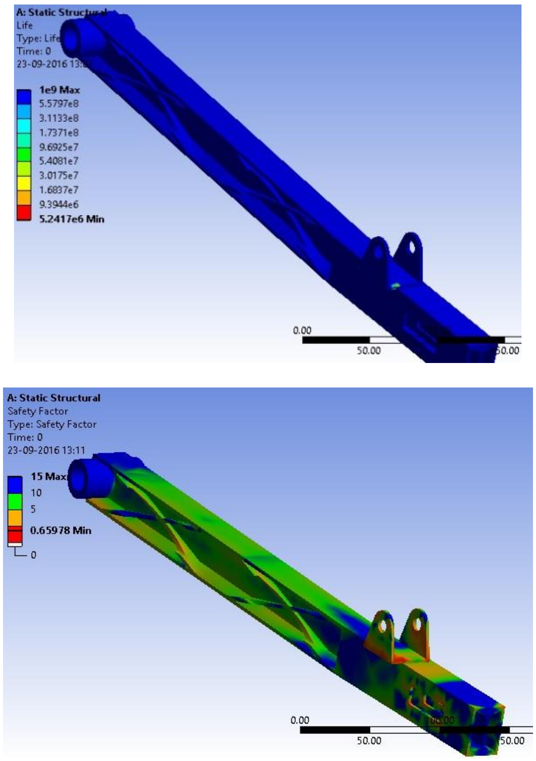


Fig.10 Life and Safety factor

The minimum life was found to be 5.2417e6 cycles whereas the minimum safety factor was found to be 0.6597. Comparatively for the original side beam, the minimum life was found to be 6.613e7 cycles and minimum safety factor of 0.73831. These parameters are somewhat less for the modified part. This problem can be fixed either by using a good surface finish or coating the part with a suitable material to increase the endurance limit of the component.

6.5 Vertical Stiffness

The value of vertical stiffness of the side beam must be less than the suspension stiffness. This is necessary to keep the seat and seat support steady (B Smith et al 2015). The FE model was assumed to be linear and during vertical loading only the maximum load of 500 N was applied. The FE strains and deflections at maximum loading were calculated and intermediate results calculated using linearity. For evaluation purposes vertical load of 500 N was applied in time step of 1 second and along with frictionless cylindrical support. The vertical deformation was evaluated to be 3.4181 mm.

Vertical stiffness of side beam K_S = 500/3.4181 = 146.28 N/mm.

Thus, the vertical stiffness was determined to be 146.28 N/mm. The rear suspension stiffness for one side of the bike is K= 102.33 N/mm. Hence, the above mentioned condition is satisfied.

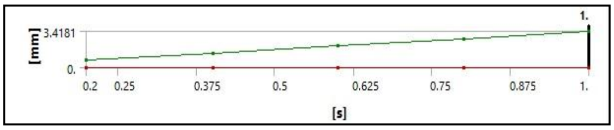


Fig. 11 Deformation vs. Time

The minimum safety factor was found to be 13.311 and minimum life was found to be 1.182e5 cycles. The maximum design life came out as 1e9 cycles indicating the infinite life design of the component. Following graph depicts the stress life relation for the component.

Conclusions

For modified swing arm, the weight was found to be 1.8 kg whereas the original swing arm weighed 3.2 kg. The stresses induced are found to be within limits. This shows that the above proposed design could be a viable option as far as weight is concerned. Further study may be needed to investigate the manufacturing feasibility. Hence the overall weight reduction achieved was 44%.

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