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Analysis and Substitution of Metal Triple Tree/Yoke of Motorcycle with Plastic Triple Tree

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Abstract

This paper is concerned with the replacement of automobile metal part with plastic part. Here the metal triple tree of motorcycle has been replaced with plastic triple tree. In this project the existing triple tree of motorcycle has been analysed and the new model has been designed using integrated analysis. The new design modelling and analysis is done on integral triple tree which have only one connecting component for clamping handlebar, other parts are integrated to achieve better mechanical reliability and performance of new model. Understanding of stress affected region in triple tree and proper analysis of the dynamic loading allow for more rapid and effective part design. A detailed design procedure is presented to optimize the torque and loads acting on this element. The dimensions of model have been optimized to achieve maximum allowable stress. The forces and steering torque have been analysed. Simulations and experimental tests are carried out in order to validate the performance of new model. The durability and performance of new designed model is parallel to the metal and alloys triple tree currently used in motorcycles.

Keywords:
Steering torque, stress, engineering plastics, integrated part.

Introduction

A triple tree/yoke attaches the fork tubes to the frame via steering head bearings, which allow the fork assembly to pivot from side to side, and therefore steer the motorcycle. Most bikes have upper and lower triple trees, providing two solid clamping points that keep the fork tubes parallel, while also connecting the forks to the frame via the steering head bearings.

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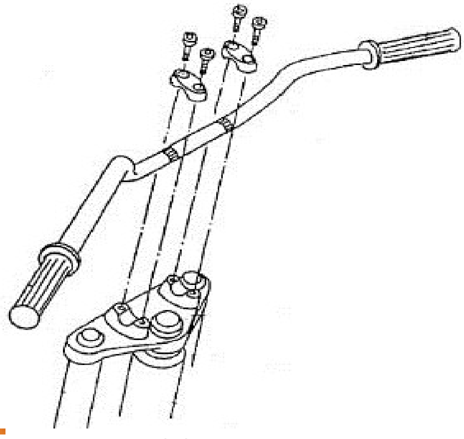
With the tendency to employ fork tubes with single action damping, fork triple trees need to be reinforced more than when the forks shared both damping roles, because the rigidity of the triple trees are relied upon to distribute the forces within the forks without flex.
The 'Telever' fork has only an upper triple tree, which allows for greater overlap of the sliders over the stanchion tubes, which improves stiffness and helps to reduce flex [10].
The existing triple tree used in most motorcycle is made of metal and its alloy having four and more connecting components. The new design modelling and analysis is done on integral triple tree which have only one connecting component for clamping handlebar, other parts are integrated to achieve better mechanical reliability and performance of new model.
Plastic components are used mainly to make automobile more energy efficient by reducing weight, together with providing durability, corrosion resistance, toughness, design flexibility, and high performance at low cost [7]. Originally plastics were specified because they offered good mechanical properties combined with excellent appearance, including the possibility of self-colouring. Weight reduction, to meet specific performance requirements and consolidating multiple metal parts into one plastic part are the primary constraints of current metal to plastic trend. The automotive industry is on brink of revolution and the plastic industry poised to play a major role. The on-going development of advanced and high performance engineering plastics, polymers and composites has dramatically increased their uses [7]. The automotive industry uses engineered polymer composites and plastics in a wide range of applications, as the second most common class of automotive materials after ferrous metals and alloys.The application of plastic components in the automotive industry has been increasing over the last decades. The increased demand of lighter automotive components while maintaining a high level of safety requires an integrated mechanical metallurgical analysis.The clamp designed here will be cheaper, easy to be processed as compared to metal clamp.
Steering torque, steering force and impact load during bump are the primary constraints for the triple tree. For the variable speed of the motorcycle, steering torque changes [1]. It is measured theoretically. This torque is necessary to be applied by the rider on the handlebar in order to maintain the desired path. The maximum steering torque at roll angle 30^0 and steering angle 20^0 is 112 N-m at motorcycle speed of 30 km/h. To have a motorcycle more maneuverability is needed that the rider feels the lowest possible force on the handlebar. So, if the torque is always same, we need to increase the distance between the axis of steering system and the point in where the rider apply force with his arms.

Problems Definition

Figure (1) shows a schematic representation of triple tree model. This design features two vertical clamps for mounting handlebar on the top of triple tree. The new design is then attempts to redesign the integrated handlebar clamps triple tree. In an attempt to integrate these clamps, these are fixed with the base plate of the triple tree. This is shown in figure (2). In order to avoid the twisting of clamps and achieving maximum steering torque in front forks during application of force on handlebar, the two ribs of height 25 mm and thickness 3 mm are provided.These ribs also provide strength to the base plate when rider apply load on handlebar. This design is depicted in figure (3a).The focus of preliminary design is to determine steering torque, load limit, stresses developed and other physical characteristics for an optimum triple tree design, which best maximizescores for competition. The complexity of the operating variable can be simplified by taking appropriate assumption. This help to develop a preliminary design. The very beginning of this project work, a model has been developed using appropriate assumption and mathematical approximation then conceptual design is feasibly represented using mathematical computer simulation software package in CATIA and Pro/E.



(a) (b)

Figure (1): Triple tree currently used

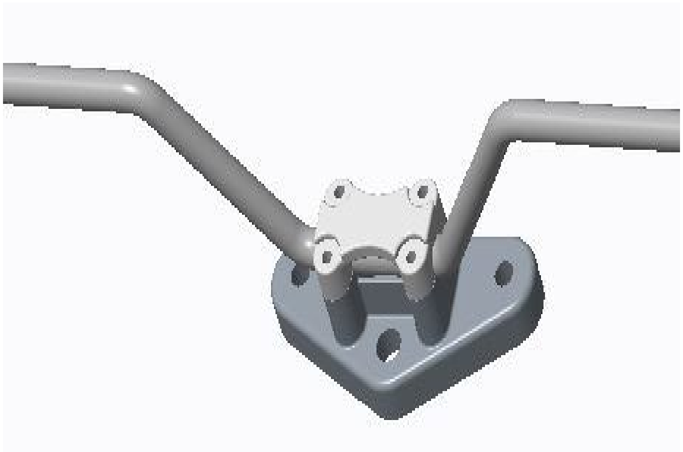


Figure (2): New designed triple tree.

Analytical Models for Triple Tree

A schematic representation of the triple tree assembly is shown in figure (2), so as to investigate its static and dynamic behaviour. Consider force *F*acting in handlebar at perpendicular distance *r* from the handlebar connector. In figure (4), a free body diagram of this acting force is illustrated. When FBD was drawn in, two considerations were taken; (I) figure (6a): since the force in handlebar apply torque in connector which induced shear stress. The shear stress at two points A and B can be calculated from the equations(6) and (7) respectively [9]. (II) figure (6b): a force F_v can act vertically which tends to bend the connector in the direction of applied force, from here bending stress can be calculate by using equation (8). The free body diagrams of these two conditions are shown in figure (6). The free body diagram of baseplate and loads acting on it is shown in figure (7)

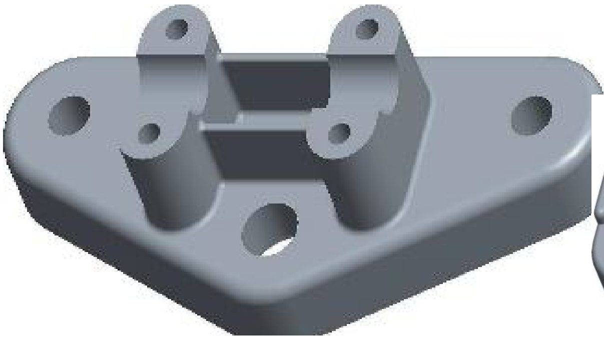


Figure (3a):Triple Tree

Figure (3b): Handlebar upper clamp.

Steering Torque

Steering torque is applied by the rider with respect to steering axis of motorcycle to maintain the desired path or turn the motorcycle. The steering torque varies with speed of motorcycle, dynamic load on front wheel, normal trail, caster angle etc. To have a motorcycle manoeuvrability, is needed that the rider should feel the lowest possible force on the handlebar[1]. So, if the torque is always same, we need to increase the distance beyween the axis of steering and the point where the rider applys force. This is shown in figure (4).

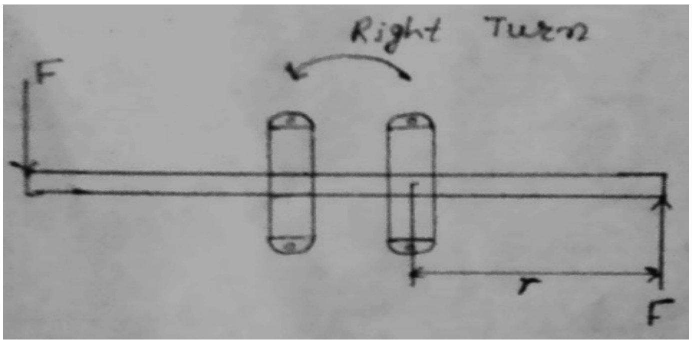


Figure (4): Steering torque on handlebar

Steering torque,

$$T = \left[F_{lf}\cos(W) - N_f \sin(W)\right]t_n \text{ -------------------- (1)}$$

where, T is steering torque, F_{lf} is lateral force induced/applied on front wheel, W is wheelbase, N_f is dynamic load on front wheel, t_n is mechanical/ normal trail.

Dynamic load on front wheel,

$$N_f = \frac{Mg}{W}(b_{cm} + \mu h_{cm}) \text{ -------------------------------- (2)}$$

where, M is mass of motorcycle with rider, g is acceleration due to gravity, b_{cm} is distance of centre of mass from rear wheel centre, μ is coefficient of friction between tyre and road, h_{cm} is height of centre of mass from ground.

Normal trail,

$$t_n = \frac{R_f \cos(\delta)\sin(\phi)}{\sqrt{1-[\sin(\delta)\sin(\phi)]^2 - O_f}} \text{ ------------------------------ (3)}$$

where, R_f is radius of front wheel, δ is steering angle, ϕ is caster angle, O_f is fork offset.

Based on calculations of above equations a graph between steering angle and steering torque at roll angle of 30^0 has been obtained and shown in figure (5). Graph shows that the steering torque increases with increase in steering angle.

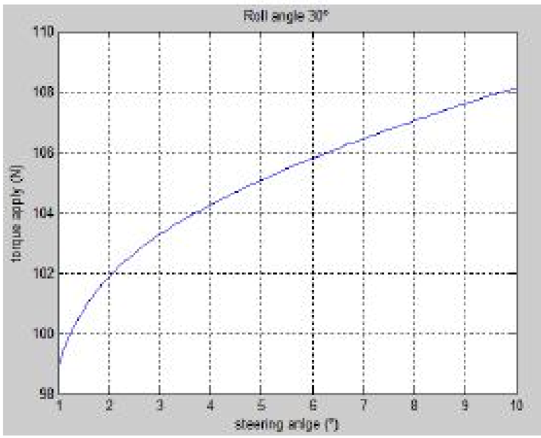


Figure (5): Steering angle vs. steering torque at roll angle 30^0

Stress in Handlebar Connectors

Due to the steering torque applied by the rider, the handlebar connectors are subjected to twisting moment. Shear stress will be induced in these connectors. Consider the connectors of rectangular cross section of length l and width b having semi-circular ends shown in figure (6a).

$$T = F.r \quad \text{-- (4)}$$

where, *F* is the force applied by rider on handlebar, *r* is distance between point of application of force and handlebar connector.

From equation (4) steering force can be calculated.

Cross section area of connector,

$$A = (l - b)b + \frac{\pi}{4}b^2 \quad \text{------------------------------------ (5)}$$

Maximum shear stress at point A,

$$\tau_A = \frac{4.81\,T}{2Al} \quad \text{-- (6)}$$

Maximum shear stress at point B,

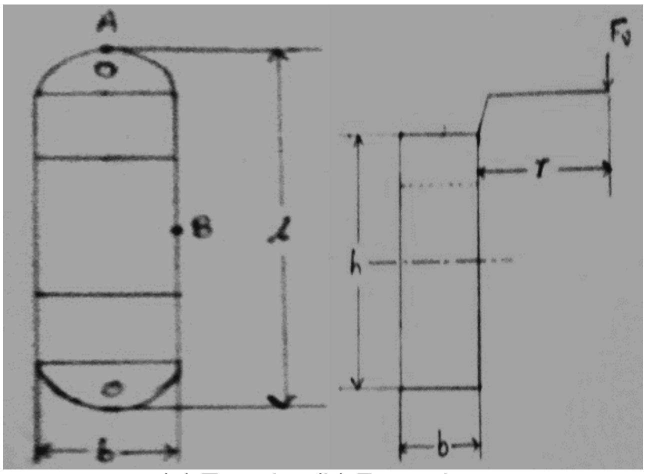
$$\tau_B = \frac{4.81\,T}{2Ab} \quad \text{-- (7)}$$

When rider sometimes applies force vertically on handlebar, it will cause bending stress in connectors which is shown in figure (6b).

Bending stress,

$$\sigma_b = \frac{My}{I} \quad \text{-- (8)}$$

where, $\sigma_b$is bending stress, *M* is bending moment, *y* is force distance from neutral axis, *I* is moment of inertia about neutral axis.



(a) Top view(b) Front view

Figure (6): Loads on handlebar connector

Load on Baseplate

When the motorcycle experience bumps on road during riding, there is an impact force induced, which is absorbed by the shock absorber. Due to this impact force, springs of shock absorber will be compressed. These compressed springs apply force on the triple trees via fork tubes. In order to find out this spring force, it needs to find out the deflection of spring or travel of forks and the load applied on the shock absorber which is roughly equal to front centre of mass of motorcycle. Considering front centre of mass 70 kg and travel of forks is 120 mm. This is shown in figure (7).

$$w = kx \quad \text{--} \ (9)$$

where, *w* is applied load, *k* is spring constant, *x* is deflection of spring.

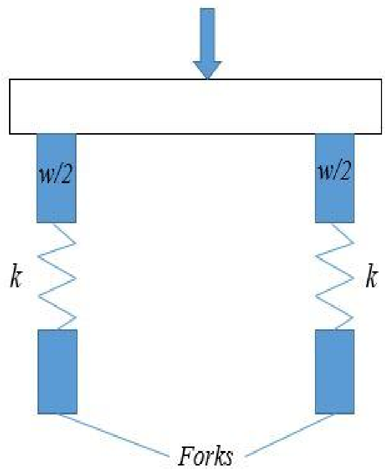


Figure (7): Loads on baseplate

Stress Analysis

Based on the above equations and calculations the stress analysis has been done in simulation software. For the analysis three conditions of loading have been adopted vs.; (i) when torque is applied, (ii) when force applied by rider on handlebar during breaking in direction of motion of motorcycle and (iii) combination of these two i.e. torque and force. The Von Mises stress at the steering torque of 112 N-m in triple tree is shown in figure (8). Notice that the stress is greater on the top of handlebar connector. In figure (9), Mises

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stress on application of force is shown and figure (10) shows combination of both. From these results one can see that the stress in handlebar connector is maximum for these three conditions upon fixing the steering head bearing hole.

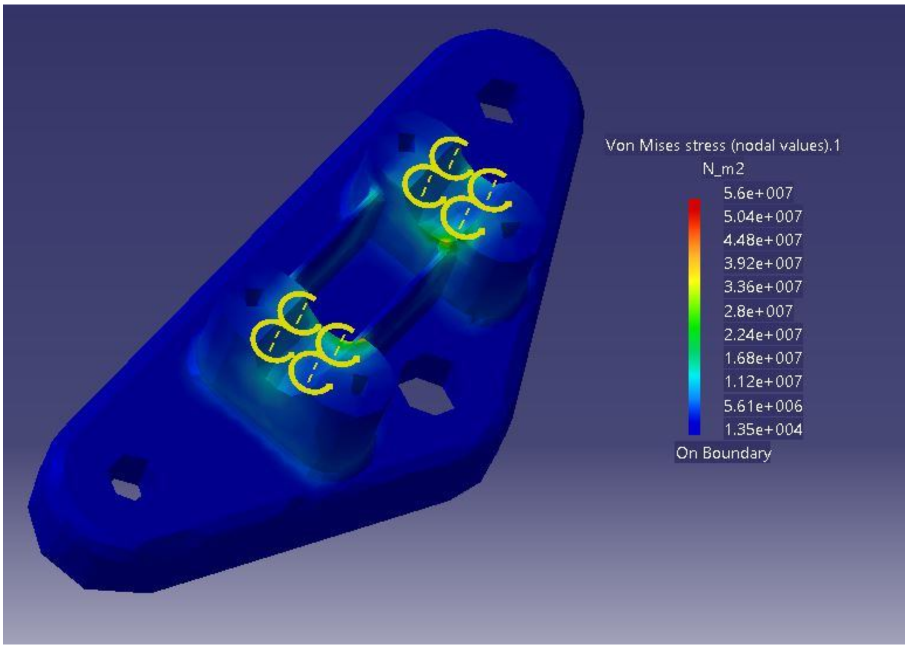

Figure (8): Von Mises stress on application of steering torque.

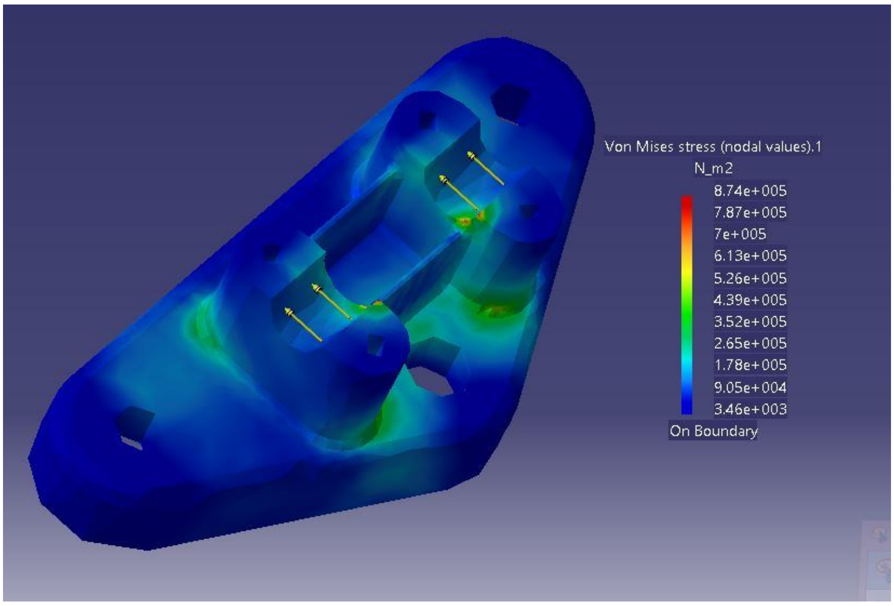

Figure (9): Von Mises stress on application of force.

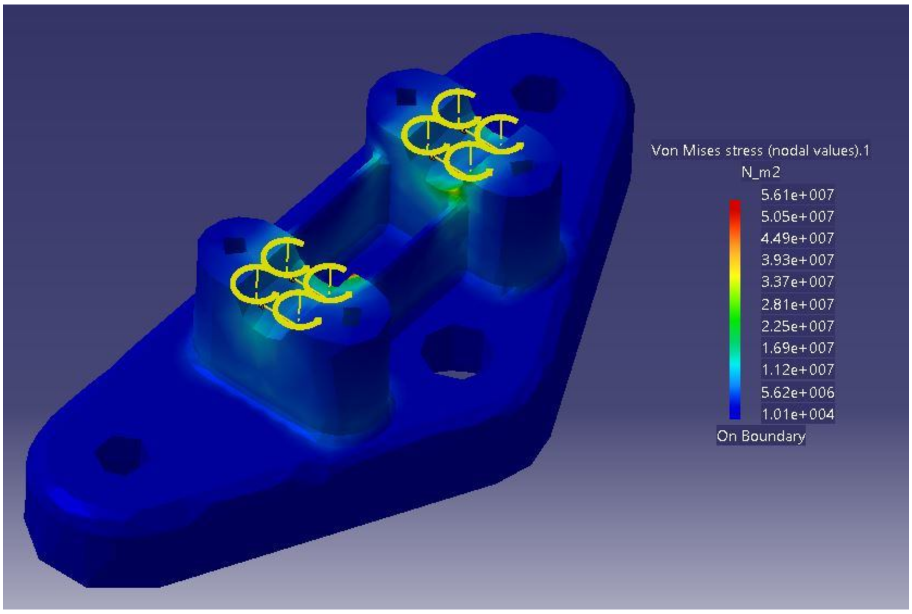

Figure (10): Von Mises stress on application of combined steering torque and force.

Optimum Design of Triple Tree

In this section, an optimum design is given for triple tree model. Maximum shear stress, bending stress, torque and impact load may be considered as the design criteria in the present problem. Variables to be optimized are thickness of baseplate, width of handlebar connector of triple tree. Based on the parameters and to simplify the fabrication first Nylon 6 is used for this model. The various design parameters are shown in table 1.

Table 1: Design parameters

Component	Symbol	Quantity	Unit
Handlebar connector	Length (l)	58	mm
	Width (b)	25	mm
	Height (h)	35	mm
	Shear stress at A (τ_A)	3.575	MPa
	Shear stress at B (τ_B)	9.431	MPa
	Torque (T)	112	N-m
Baseplate	Overall length	220	mm
	Overall Width	92.5	mm
	Offset (O_f)	42.5	mm
Others	Wheelbase (W)	1255	mm
	Front wheel radius (R_f)	431.8	mm
	Caster angle (ϕ)	26.5	degree
	Steering angle (δ)	20	degree
	Spring deflection (x)	120	mm
	Front centre of mass (w)	70	kg

Conclusion

In this project, the triple clamp has been redesigned to provide better mechanical reliability and performance. This has been achieved by replacing the connecting components with integral connectors. The dimensions of the triple tree have optimized to provide maximum allowable stress. Different types of load acting on the triple clamp through handlebar and fork has analysed and stresses are calculated, the maximum shear stress in handlebar clamp is 9.5 MPa at a torque of 112 N-m.The forces and torque have been analysed and compared with metal part. The design is validated in a digital simulation and mathematical analysis. The durability and performance of new design is parallel to the metal and alloys triple clamps currently used in motorcycles. The present redesigned model of plastic triple tree is as reliable as metal triple tree.

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