# DESIGN AND DEVELOPMENT OF MOTORCYCLE CHAIN LINKS BY USING C.A.E. SOFTWARE

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## ABSTRACT

*The new motorcycle to be launched in the automobile market needs to be ensured for safety and efficiency. Chain drives being efficient means of power transmission are preferred for this product. The limitation, of course, being catastrophic failure at virtually no prior notice. At high speed, accidents are very likely in case of failure in the chain link. The problem needs to be looked into and investigated for identifying causes for failure. Similarly, upcoming variants to be incorporated with the new solution to eliminate such challenges in the future. The design for the chain would be subjected to F.E Analysis as an Analytical Methodology to find the effect of loads (tension) on the link. The link being a `unit' of the existing chain would be assessed for performance while tensile loads are exerted at both its ends. Safe loads would be determined and the design tested for safe use in the Automobile. The problem for this work being evaluation of the design using Analytical methodology followed by experimentation to validate it. An existing chain link would be used for benchmarking the research work. Finite Element Analysis tools like HyperMesh and ANSYS are suitable to find the performance of the link under tensile loads. Recommendation over the best suited geometry or material would be presented to conclude the work.*

***KEYWORD:*** *Analytical Methodology, ANSYS, Chain, Chain link, tensile loads.*

## I. INTRODUCTION

Chain drives offer non-slip, light weight, inexpensive, compact power transmission compared to belt or gears, but usually at the cost of increased noise and vibration. A brief history of chain drives and the important milestones in their practical development through the late twentieth century can be found in Conwell

All chains 'stretch' during their lifetime and eventually need replacing. Chains don't stretch in the same way elastic bands do – they get longer because the metal in the links gradually wear away and makes the overall length of the chain increase. As the chain stretches, the amount of free play increases and you eventually have to move your rear wheel back a bit to take up the slack. If there's too much slack, the chain will jump around lots whenever you change speed. If there is too little slack, the chain will get over tensioned when you slow down and the back end of the bike becomes un weighted. When you replace the chain, always replace the sprockets too – they're much cheaper than the chain anyway. It's a false economy not to, since putting a new chain over worn sprockets will make your chain wear out faster. It's much easier to loosen the bolt which holds the front sprocket when the chain is still on the bike. You put the bike into a high gear and get someone to stand on the rear brake while you loosen the holding bolt a bit. Since the front sprocket is still attached by the chain to the rear sprocket, it can't spin around while you try to loosen it.

Roller chain, which is the ultimate in chain design, and constitutes the majority of chain produced today, is a relatively new invention. Now they have achieved the highest levels in the world for both quality and quantity. This holds true for conveyor chain, as well. The industries that are the main users of the chain, including automobile, electronics, steel, chemical, environmental, food, bicycle, and motorcycle industries, have developed new technologies and production methods that require various high performance chain. These industries are looking for improvement in tensile strength, fatigue strength, abrasion resistance, environmental resistance, and efficiency, as well as perfection of maintenance-free chain products. To satisfy these many requirements, chain makers are making every effort to improve chain's basic performance step by step.

## II. VEHICLE SPECIFICATION

**Table 2.1** Inputs Data for TVS 250CC Motorcycle.

| Specifications of TVS 250CC | |
|---|---|
| Engine Type | Single Cylinder, 4 Stroke |
| Engine Displacement (CC) | 250CC |
| Maximum Power | 22kW@9500 |
| Maximum Torque | 20Nm@8000 |
| Transmission | 4-Speed Constant Mesh |
| Weight | 140kg |
| Tyre size front | 100/80 -17 |
| Tyre size rear | 110/70-17 |
| Primary Reduction | 2.93 |
| 1st Gear | 1.55 |
| 2nd Gear | 1.30 |
| 3rd Gear | 1.21 |
| Top Gear | 0.83 |
| Final Reduction | 2.93 |
| Drive sprocket no of teeth , Z1 | 15 |
| Driven Sprocket no of teeth, Z2 | 44 |
| Chain Pitch, p | 12.7 mm |
| Max. Speed of smaller sprocket, n1 | 5000 rpm |

## III. CHAIN FORCES IN TIGHT SPAN

Based on the power at drive wheel, it is found that under full throttle condition maximum tractive force is developed and this would cause more force to act on transmission chain. Hence, in this present investigation, forces on chain are calculated for motorcycle of brand B, model 1 for maximum power condition. Here, vehicle acceleration is classified into two stages. In stage one, vehicle starts under first gear and during this stage, maximum torque is required to accelerate with pay load and the acceleration of vehicle continuous through $2^{nd}$, $3^{rd}$ till it reaches considerable speed. In stage two, vehicle is further accelerated in top gear by giving more throttles, till vehicle attains maximum steady speed overcoming all resistances. Motorcycle operates mostly under maximum torque condition and in stage two it operates under maximum power condition. Hence, in this research work, for calculation of chain force, maximum torque condition of engine is taken for stage one and maximum power condition is taken for stage two. Also, efficiency of primary reduction and gear box reduction, which normally varies from 95% to 96%, is not considered in the present study so that it accounts for maximum force acting on chain. Therefore, without considering the said efficiency, using engine and transmission specification from Table. In this theoretical evaluation work, it is assumed steady state condition and hence chain forces are taken under maximum power condition under full throttle with the assumption that motorcycle runs at maximum speed continuously with zero acceleration. However, to study the effect of speed variation from stalled position to different acceleration and braking, chain forces and speeds are taken both from maximum power and maximum torque conditions under full throttle.

$$Pe = \frac{2\pi Nc}{60000}$$

where $P_e$ engine power in kW
$N_C$ rpm of crankshaft.

**Table 3.1** Force on chain under maximum engine power condition

| Description | I Gear | II Gear | III Gear | IV Gear |
|---|---|---|---|---|
| Drive sprocket speed rpm | 1984 | 3075 | 3998 | 4798 |
| Driven sprocket speed rpm | 677 | 1049 | 1365 | 1637 |
| Driven sprocket torque Nm | 310.32 | 200.27 | 153.08 | 128.36 |
| Force on chain at max power N | 3351.55 | 2162.982 | 1653.31 | 1386.33 |

**Table 3.2** Force on chain under maximum engine torque condition

| Description | I Gear | II Gear | III Gear | IV Gear |
|---|---|---|---|---|
| Drive sprocket speed rpm | 1670 | 2588 | 3364 | 1402 |
| Driven sprocket speed rpm | 570 | 833 | 1148 | 1378 |
| Driven sprocket torque Nm | 368.57 | 237.92 | 183 | 152.46 |
| Force on chain at max torque N | 3980.66 | 2569.61 | 1976.45 | 1646.61 |

# IV. FINITE ELEMENT ANALYSIS

## 4.1 Basic Steps followed in present to study.

### 4.1.1 CAD Model Generation

Two wheeler transmission roller chains are chains with 12.7 mm pitch. Motorcycle chains, which are being used in existing motorcycles, pin diameter and inner plate thickness compared with regular 12.7 mm standard chain used in Motorcycle. shows structure of roller chain as per chain specification of chain part.CATIAV5 were used to create model.

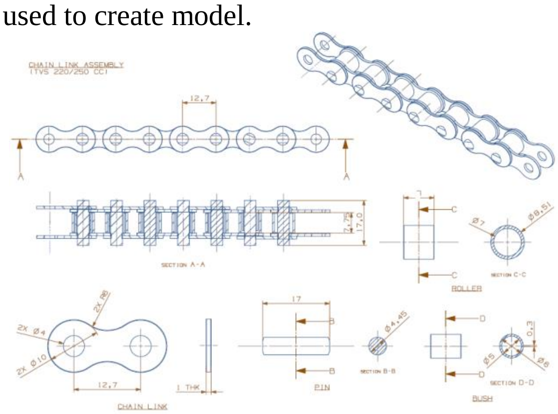
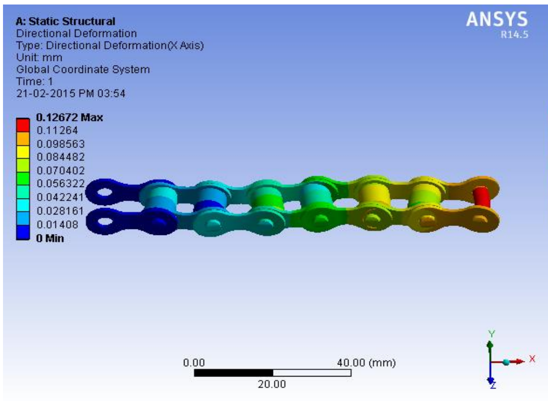


**Fig.4. 1** Three Dimensional Model

## 4.2 Finite element analysis

In ANSYS the Cad Model of Chain link is developed. After that for analysis the Finite element model is generated in engineering data define defining material properties. In that entering the physical and material properties of the model to the software, defining the element type meshing criteria, Mesh body consisting of total solid body is created. After modeling the chain link meshing is done in ANSYS Workbench. Meshing involves converting of geometry into nodes and elements. Supporting boundary conditions are applied i.e. fixed support and horizontal support. In fixed support there is no any degree of freedom i.e. there is no displacement at any direction. The FEA results of Chain link for Direction deformation and Equivalent (von-Mises) Stress as calculated by using C.A.E. In Fig. 4 red face indicate maximum deformation occurs on pin which is 0.12672 mm and blue face indicate minimum deformation.



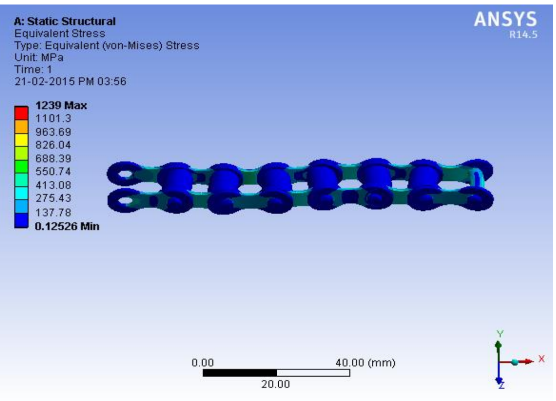




**Fig.4.2** Deformation in Model **Fig.4.3** Stress Distribution in Model

The stress distribution in chain link having maximum value of stress that is 1239Mpa at Roller and minimum value of stress that is 0.12526MPa at Pin. Maximum stress on pin compare to yield strength pin material which that give stress is more than yield strength of material due to this this chain not used for 250cc Motorcycle.
The Chain design is not safe for the given loading conditions. The stress values are not within the allowable stress limits

## V. The Different Cases To Carried Out For The Evaluation Results By Finite Element Analysis

The different theoretical cases are to carried out for the following configurations and evaluation of these results by finite element analysis -

a) Seamed bush v/s seamless bush
b) At change Design Parameters Inner Link and outer Link ,Roller
c) Thickness of Inner Link and Outer Link increase by 0.25 mm
d) Non-linear

### 5.1 Seamed bush v/s seamless bush.

Currently the bush used in current chain drive is manufactured by rolling a flat strip, hence it had a seam in between as shown in Fig. The place as where did seam got fitted in the assembly was studied by changing the angle. Finally comparison was made in between the seamed bush and seamless bush.
While assembling the chain, there is no particular orientation of the bush. The seam in the bush gets fitted at any angle. An attempt to simulate the different orientation of the seam and the corresponding stresses was made. Seamless bush gave the minimum contact stresses and is hence preferred.

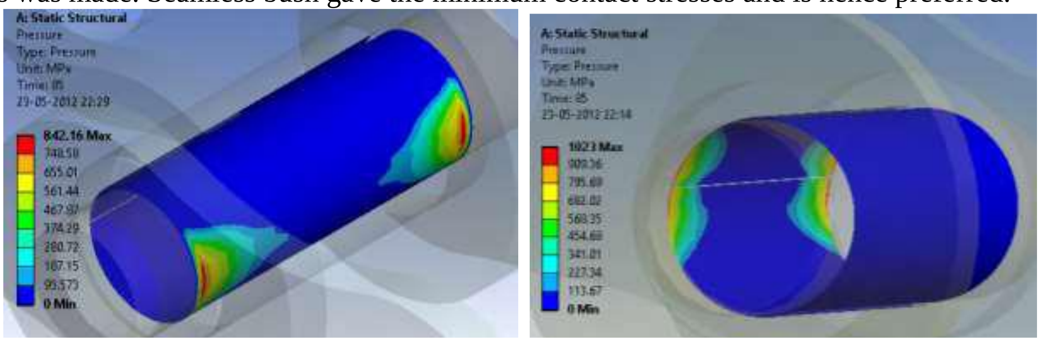
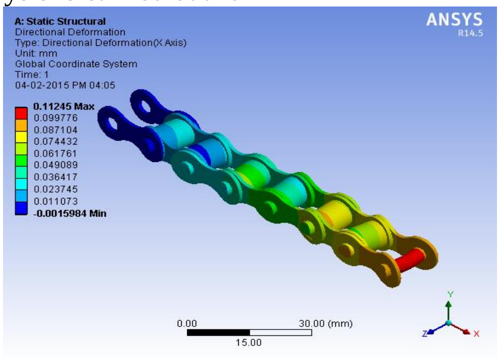



**Fig. 5. 1** Result for seam less bush used

### 5.1.1. Iterations For used seamless bush,

Here analysis is carried out for -



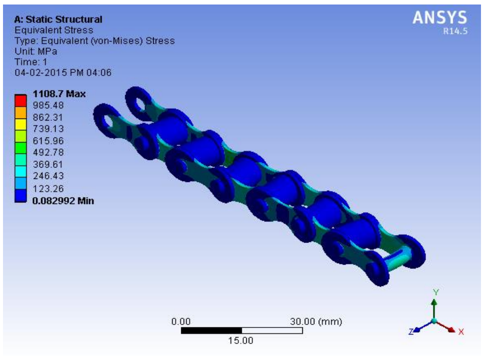




**Fig. 5.2** Deformation **Fig. 5.3** Stresses : Von Misses Stress

## 5.2 Iterations For At change Height Inner Link plates and Outer Link plates , Width Roller for 1mm thickness

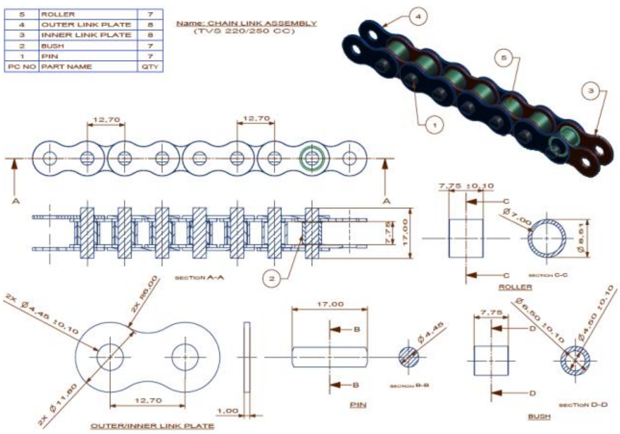


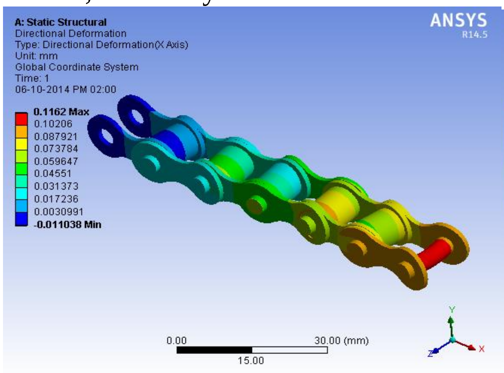

**Fig.5.4** Modified Three Dimensional Model for iteration No.2

### 5.2.1 Structural Static Analysis

Analyze of the structural strength & deformation of the chain against maximum force on chain is carried out.

### 5.2.2 Static Analysis Results

The analysis of traditional slat conveyor is carried out by using following inputs-
Material data ,The Analysis Results Includes: Deformations ,Stress



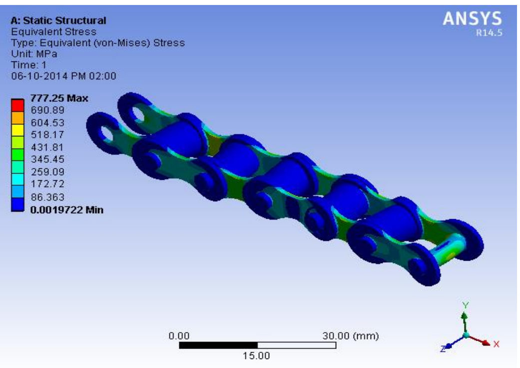




**Fig.5.5** Deformation (Overall) **Fig.5.6** Stresses : Von Misses Stress Plot

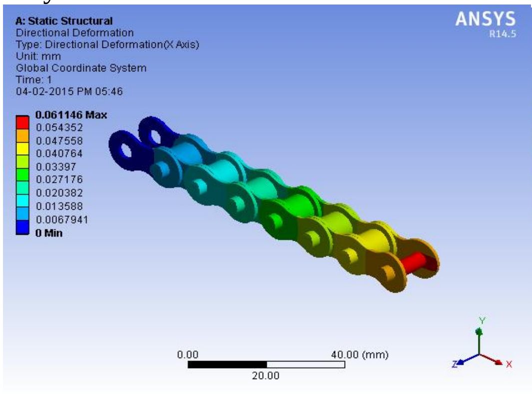
**Results:** The maximum deformation of slat is 0.1162 mm which is more. The maximum stresses in slat is 777.25 MPa which is more.

**Conclusion:** The Chain design is close to safe for the given loading conditions.

The stress values are not within the allowable stress limits

## 5.3 Iterations For Inner link and Outer link Thickness = 1.25mm ,

Here analysis is carried out for -



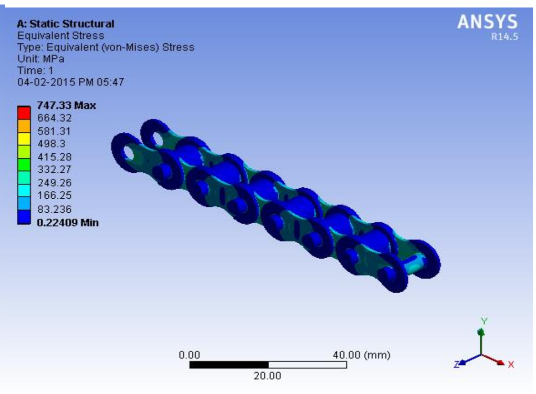
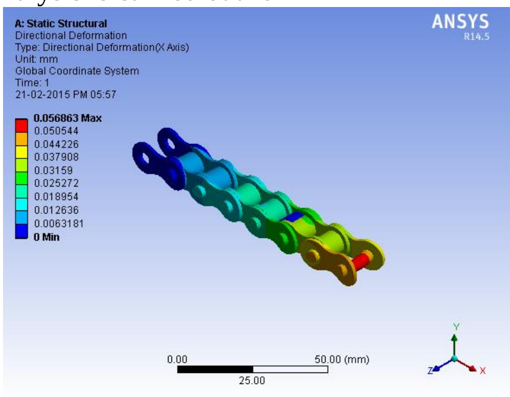




**Fig. 5.7** Deformation (Overall) **Fig. 5.8** Stresses : Von Misses Stress

**Results:** The maximum deformation of slat is 6.1146e-002 mm which is very less.
The maximum stresses in slat is 747.33 MPa which is very less.
**Conclusion:** The Chain design is close to safe for the given loading conditions.
The stress values are not within the allowable stress limits

## 5.4. Iterations For Inner link and Outer link Thickness = 1.5mm ,

Here analysis is carried out for -



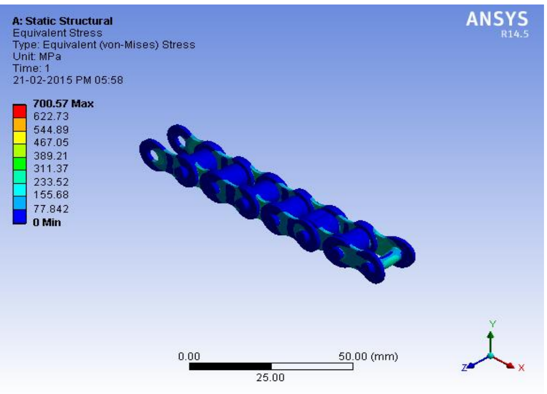
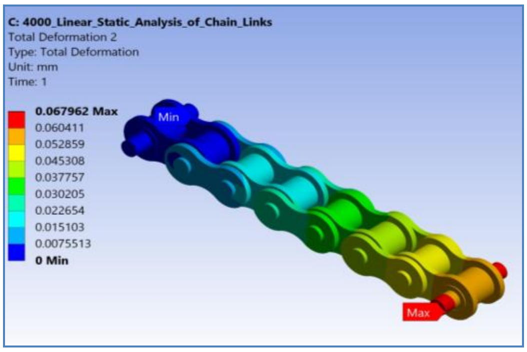




**Fig. 5.9** Deformation **Fig. 5.10** Stresses : Von Misses Stress

**Results:** The maximum deformation of slat is 5.6863e-002 mm which is very less. The maximum stresses in slat is 700.57 MPa which is very less.
**Conclusion:** The Chain design is close to safe for the given loading conditions. The stress values are within the allowable stress limits

## 5.5. Iterations For Changing Profile of Inner link and Outer link Thickness = 1.5mm



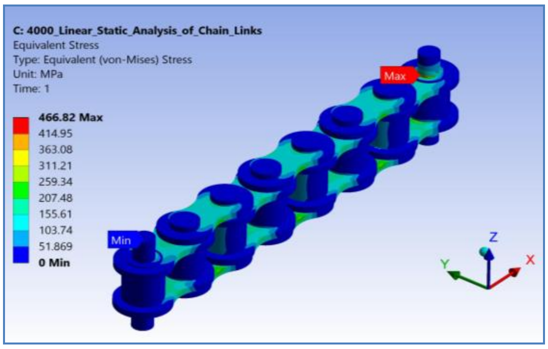
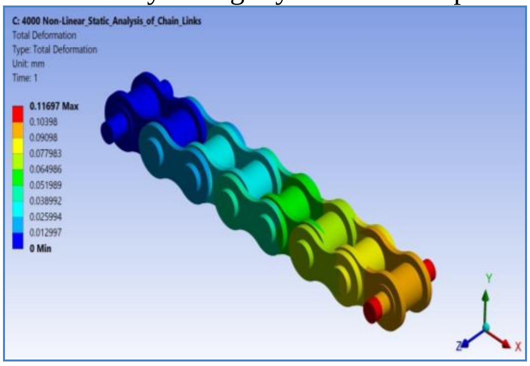
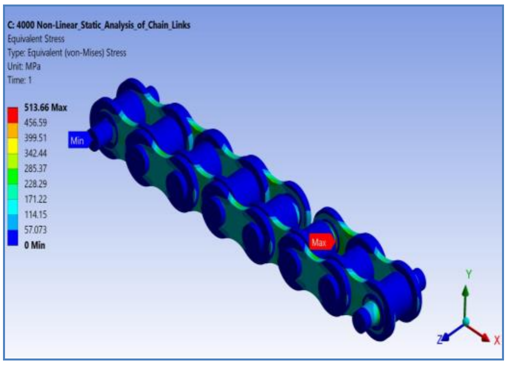




**Fig. 5.11** Deformation **Fig. 5.12** Stresses (Slat): Von Misses Stress

## 5.6 Iterations For Nonlinearity for chain link Thickness = 1.5mm

In geometric nonlinearity we have considered effect of large deflections and in contact nonlinearity we have considered the actual real life contact scenario of system. That is there is standard contacts between rollers and links which means there is only sliding can occur in these two bodies no penetration or no load transfer will occur. Similarly in case of two links both are sliding against the o ring where as they are rigidly connected to pin that is outer link and inner link to bush.




## Conclusion

The Chain design is close to safe for the given loading conditions. The stress values are within the allowable stress limits

**Table 2** Results of F.E Analysis

| **Design Parameter** | **Maximum Tensile stress, (MPa)** | **Maximum deformation(mm)** |
|---|---|---|
| **Existing Chain** | **1239** | **0.12672** |
| **Seamless bush** | **1108.7** | **0.11245** |
| **Changing Profile** | **777.25** | **777.25** |
| **Thickness 1.25** | **747.33** | **0.061146** |
| **Thickness 1.50** | **700.57** | **0.056863** |
| **Thickness 1.50** | **643.1** | **0.053343** |
| **Changing Profile** | **466.82** | **0.067962** |
| **Non linear** | **513.66** | **0.11697** |

# VI. CONCLUSION

The design for the chain would be subjected to F.E Analysis to find the effect of loads (tension) on the link. The proposed method utilizes software in the FEA domain for analyzing the effects of the variation in the values of the design parameters influencing the performance criterion. As the seamed bush and seamless bush, change design parameters of inner links and outer links, roller, bush, thickness of increases. If the thickness of link plate increase to the rate of decreases in tensile stress. Consequently weight with a increases in the thickness of the link plate can be realized using a higher strength of material. The FEM method is used to analyze the stress state of an elastic body with a given geometry, such as chain link.

Based on present work of developing model for tensile test, the following conclusions are drawn:

1. Seamless bush is found superior over the seamed bush and can be used in the new chain manufacturing. Modified bushes may cause a small bush projection over inner links due to material flow, which may help in reducing the lateral movements of links making the chain move in an almost straight path than swaying sideways.
2. Contact non-linearity is successfully simulated with the help of frictional contact. Non-linearity and the stress, strain, deformation variation over the model can be successfully observed.
3. Based on the proposed dimensions, a new model has been proposed with high tensile strength than the proposed one

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