

Weight Optimization Using Topological Approach and Strength Evaluation of Chain Link Plate

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ABSTRACT: In industries such as sugar factories where conveyor system are used, power consumption is more due to bulky stacker chain. So, to reduce power consumption we have to reduce weight of chain link plate by using topological approach. The material used for chain link plate is EN19 for better strength. By using ANSYS workbench shape optimization of industrial chain has been carried out to remove material of the chain where stresses are low for applied loading and boundary condition. Shape optimization gives proper shape of chain link plate. We got weight reduction up-to 15% for redesigned chain. Structural analysis is carried out by using ANSYS workbench to find out maximum stresses in the chain link plate. Experimentation is carried out on redesigned chain link plate by using computerized universal testing machine to find the breaking load of the chain link plate. The breaking load is in the range of +/- 10 % of initially working load. So, the new design is safe for application of working load. By experimentation it is noticed that failure occurred where stresses are maximum in chain link plate.

KEYWORDS: ANSYS workbench, breaking load, shape optimization, structural analysis, topological approach.

I. INTRODUCTION

Conveyor chain is the major mechanical material handling system which is used in today's industry. So, it is very important to study about its failure and try to save it from failures. In case of long distance transmission with metal chain, large driving force is required to cover the huge mass of chain that means power requirement is also high according to the weight of chain. So, weight is the main problem with metal chain. Moreover, most of the time chain is under tension while moving on track with heavy pallets and this tension which causes failure of chain assembly which is the major problem for industrial sector such as, wear, fatigue failure, noise, vibration, pin bushing galling etc. Causes of this failure are improper design.

Topology optimization starts with an initial design (the original design area), which also contains any prescribed conditions (such as boundary conditions and loads). The optimization process determines a new material distribution by changing the density and the stiffness of the elements in the initial design while continuing to satisfy the optimization constraints, such as the minimum volume or the maximum displacement of a region. Ansys can apply the following objectives to a topology optimization process such as Strain energy, Eigen frequencies, Internal and reaction forces, Weight and volume, Centre of gravity, Moment of inertia. The objective of a shape optimization is to minimize stress concentrations using the results of a stress analysis to modify the surface geometry of a component until the required stress level is reached. Shape optimization tries to arrange the surface nodes of the selected region until the stress across the region is constant. Only a volume constraint can be applied to a shape optimization. In addition, a number of manufacturing geometric restrictions can be applied that ensure the proposed design can continue to be produced using casting or stamping processes by freezing selected regions and apply member size, symmetry, and coupling restrictions.

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For instant now material steel EN8 have been using for link. But to reduce the weight of stacker chain we can use topology optimization that is shape optimization. We have to redesign the chain using the CAD software and topology optimization should be done in Ansys Workbench tool. The basic problem statement is to design the stacker conveyor chain to reduce the weight by using topological concept.

II. LITERATURE REVIEW

Ledvina and Hummel [1] have developed a randomized sprocket for roller chain. After studying this patent it is noticed that due to conventional random sprocket mechanism failure of the chain occurs due to the wear. So, we can use roller chains to avoid wear in the chain during conveying process. P.R. Payet [2] invented conveying method to convey the material at required speed. There various conveying methods that have numerous disadvantages, as the thick layer of cane is not suitable for saturating with water or juice, and also, the very low speed imparted, equivalent to a low relative flow, requires heavy and expensive feed lines both for conveying and transmission. The purpose of the invention is to produce a process enabling sugar cane to be conveyed at the required speed in a mill train. Another purpose of the invention is to produce a device enabling very high conveying speeds to be obtained and a regular feed to the three-roller crushing mills. Another purpose of the invention is automatically to rid the sugar cane of the tramp iron that may be found therein. The purpose of the invention is to impart to the crushed or chopped cane 'a speed independent from that of the mills, which can be as high as the user requires, by means of a motor independent from the mill, endowed with a 'given power. This motor drives a rotor which propels the bagasse. From his invention it is noticed that system is very bulky due to conventional design. So, if we use topological shape optimization to get proper shape then we can reduce the weight of the conveying system and ultimately we can achieve the more speed to convey the material for his mechanism. Bhoite et al. [3] studied FEA based effect of radial variation of outer strip in a typical roller chain strip assembly. They summarized various design variables, such as wall thickness of strip, breaking area of strip and shape of the strip to formulate an idea of the system. From this paper it is noticed that topological optimization can be done using FEA tool like ANSYS Workbench and it will gives best result for given boundary condition. Sapate et al. [4] had done metallurgical investigation of fractured connecting pins of drag chain conveyors. The failure analysis of two fractured pins was carried out. The metallographic studies showed non-uniformity in the induction hardening and undesirable coarse martensitic microstructure at the core. The analysis of the fractured surface by SEM indicated fatigue fracture due to bending stresses and mild to moderate torsional stresses. The proper induction hardening to ensure required case depth and use of En-24 steel for connecting pins has been suggested to further improve life of the connecting pins. If we done hardening process for particular period (30-60 minutes) by maintaining definite temperature rate in the austenite phase then the properties of the material gets improved. Then the material becomes more resistive to fatigue fracture. M. Sujata et al. [5] studied failure of engineering components due to presence of defects in the material. These defects are either present in the material from the casting stage or get developed during subsequent hot working and thermal treatment operations. Identification of the origins of defects is an important task while analysing failures where pre-existing defects in the material are the causative factors. It was determined that the failure was caused by defects related to the metal processing. These defects were identified as surface defects in the billet, which got translated into lap or fold like defects in the final products. It was recommended that the billet be properly dressed and the surface defects are removed prior to forging operations. From this paper it is noticed that heat treatment process can be used to decrease the defects in the material and improve the material properties like tensile strength. Jagtap et al. [7] studied the experimental demonstration of strain gauge on chain strip in a Strain Rosette - 45° manner. Finally, experimentation is carried out on Computerized Universal testing Machine (UTM). This work concludes that with increase in load there is linear behaviour of strain in loading direction. There is inversely relationship up to around 90KN, followed by exponential decreasing behaviour in perpendicular to loading direction. Whereas for 45 inclined strain gauge there is linear decreasing behaviour up to 85KN later followed by exponential growth. Jagtap et al. [8] investigated the behaviour of chain strip under tensile loading. In turn it will help in reducing down time and maintenance cost related to chain assembly in various industries. In this paper it is noticed that by doing heat treatment of material we get the higher tensile strength of the chain link plate.

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III. OBJECTIVES

1. To do theoretical estimation of required breaking load for chain link plate.
2. Shape optimization of chain link plate using ANSYS Workbench 15 software and redesign of chain link plate.
3. To find, the strength of redesigned chain link plate by conducting tensile test on universal testing machine.
4. Comparing results of theoretical, Finite Element Analysis and experimental.
- 5.

IV. BREAKING LOAD CALCULATION

As per the ASME B29.1M standards [6] I have taken chain link plate of EN-19 material, dimensions 68.30 mm x 76.20 mm (Pitch) x 9.5 mm for chain link plate.

The thickness of link plates for the standard series is,

$$\therefore \text{Thickness of plate} = \frac{1}{8} \times \text{Pitch} \dots \dots \dots (1)$$

The maximum height of link plate is,

$$\therefore \text{Height of plate} = 0.82 \times \text{Pitch} \dots \dots \dots (2)$$

TABLE I
MATERIAL PROPERTIES

Sr. No.	Parameter	Descriptions
1	Material	EN19
2	Young's Modulus (E)	$2.05 \times 10^5 \text{ N/mm}^2$
3	Density (ρ)	$7.800 \times 10^{-6} \text{ Kg/mm}^3$
4	Poisson's ratio	0.3
5	Tensile Strength	1231 N/mm^2

Now by using the analytical formulae we find out the value of maximum stress i.e. ultimate tensile strength.

$$\text{Tensile strength} = 1231 \text{ N/mm}^2$$

$$\text{Modulus of elasticity} = 2.05 \times 10^5 \text{ N/mm}^2$$

$$\text{Poisson's ratio} = 0.3$$

$$\therefore \text{Maximum Working stress} = \frac{\text{Maximum stress}}{\text{Factor of safety}} \dots \dots \dots (3)$$

$$\therefore \text{Maximum Working stress} = \frac{1231}{1.5}$$

$$\therefore \text{Maximum Working stress} = 821.45 \text{ N/mm}^2$$

So, as per the above analytical calculations I got maximum working stress of 821.45 N/mm^2 . Now by using the working stress values calculate the working load the chain plate can carry by using following formulae.

Maximum working load for outer link plate is,

$$\therefore \text{Maximum working stress} = \frac{\text{Working load}}{\text{Area perpendicular to tensile load}} \dots \dots \dots (4)$$

$$\therefore 821.45 = \frac{\text{Maximum Stress}}{2 \times (17.285 \times 9.5)}$$

$$\therefore \text{Maximum working load} = 821.45 \times 328.415$$

$$\therefore \text{Maximum working load} = 269.77 \text{ KN.}$$

From above calculation we got the maximum working load 269.77 KN for chain link plate.

V. SHAPE OPTIMIZATION OF CHAIN LINK PLATE USING FEA

In this work ANSYS Workbench is used to find the optimum shape of the link plate. This software comes with Shape Optimum Module. Initial and last concept of optimization is shown in this paper.

As per ASME Standard baseline design model is considered as shown figure 1 for an initial concept. Within the link, most dimensions in the industry are parametrically defined, however one dimension, the radius that is in between inter connecting holes is left to manufacturer convenience.

I. Initial Concept

In initial concept I have considered rectangular plate of fixed thickness 9.5 mm and pitch 76.20 mm. There are basically three steps for finite element analysis explained as,

1. Pre-processing:

In Pre-processing CAD model is created using CATIA V5 R20 software for initial concept and imported in ANSYS Workbench 15. Shape optimisation module is used to get optimised design of chain link plate. After that meshing, constraint, mechanical properties and loads are applied to the chain model.

2. Analysis:

In analysis or processing result is calculated using solver.

3. Post-processing:

In post-processing result plot are shown such as shape finder, stresses, displacement for all iteration.

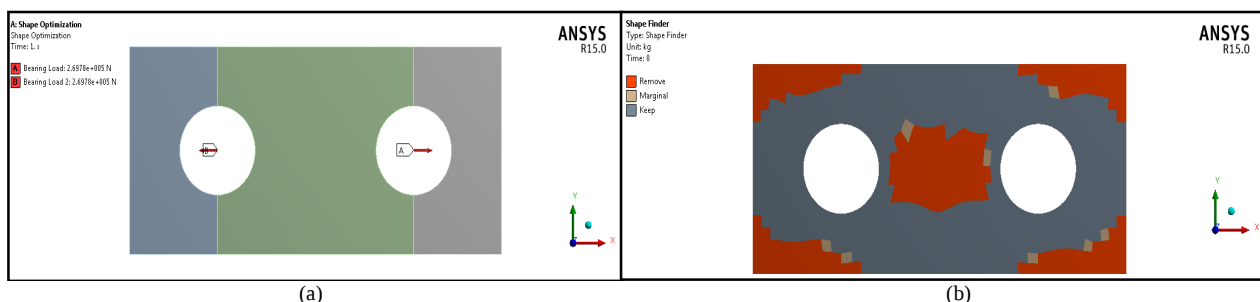


Fig.1. (a) Boundary conditions on initial concept and (b) Shape finder of chain link plate initial concept

For all iteration same process is followed. We have shown initial and final iteration of optimisation of chain link plate. Figure 1(a) shows the Concept-1 with applied boundary conditions on that. Concept-1 design for the link plate is shows the bearing load is applied at the pin-hole interface. Shape finder shows the material which can be removed from the initial model due to low stress region.

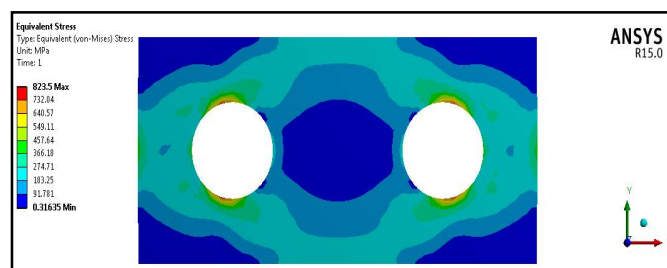


Fig.2. Von misses stress plot of initial concept.

From Equivalent Stress plot as shown in Figure 2 and shape founder plot as shown in Figure 1(b), the next iteration of out model comes. Corners of the plates are removed from link plate referring the contours of the low stress regions and shape finder plots.

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II. Last Concept

Final concept shows evolved model through history followed by engineering statistics to gain required link model as per standard topology optimization process. Shape finder shows that there is very less material has to be removed from the chain model.

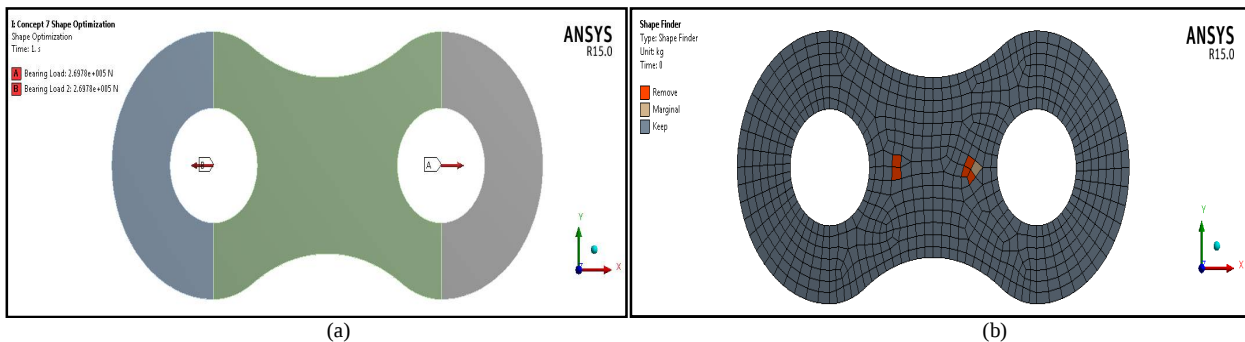


Fig.4: (a) Boundary conditions on last concept and (b) Shape finder of chain link plate final concept.

Figure 4(a) shows the last concept with applied boundary conditions on that. In figure 4(b) last concept design for the link plate is shows the bearing load is applied at the pin-hole interface. Shape finder shows the material which can be removed from the initial model due to low stress region.

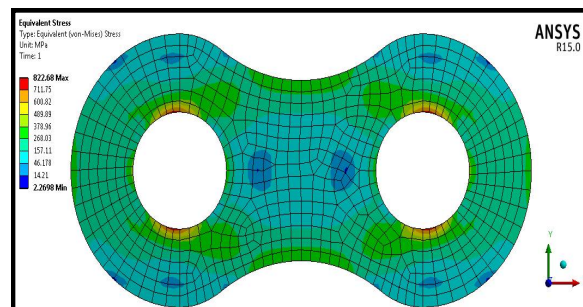


Fig.5. Von misses stress plot of final concept.

As shown in Figure 5 red colour indicates there is maximum stress concentration near to hole in the outer link plate; therefore chances of failure are near the hole. At a maximum load of 2, 69,775 N it can sustain maximum stress of 822.68 N/mm². Weight of link plate is 0.504 kg and after shape optimization weight is reduced to 0.431 kg. So, there is 15 % reduction in weight of the link plate.

VI. EXPERIMENTAL DETAILS

For validation of analytical and numerical results we need to test the specimen on Computerized Universal testing machine and compare the result of analytical, numerical and experimental result. To get optimized shape of link plate wire-cut electric discharge machining (WEDM) is used for manufacturing process. Work piece is cut by a single metal wire in the tank filled with dielectric fluid in wire-cut EDM process. Dielectric fluid is used as coolant while cutting process and removes the waste material. This process used for complicated and delicate shapes. This process is suitable where less residual stresses are required. In this process plate up to thickness 300 mm can be cut. Mostly, water is used as dielectric fluid because we can control its resistivity and other various electric properties using filters. Two fixtures made up of EN24 steel material are used to hold the link plate to perform loading test on universal testing machine.

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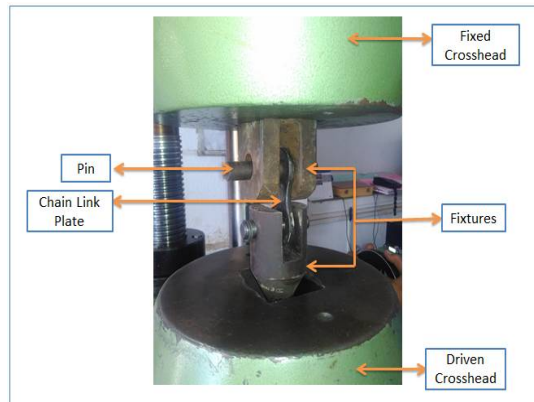


Fig.6. Experimental set-up

As shown in above Figure 6 the chain was clamped on the Universal Testing Machine with the help of two fixtures and pins. The material used for fixtures and pins is EN24. The load was given by the application of the hydraulic pressure. The breaking load of EN19 link plate is found to be 290 KN. The graph of load versus displacement is shown in figure 7(a).

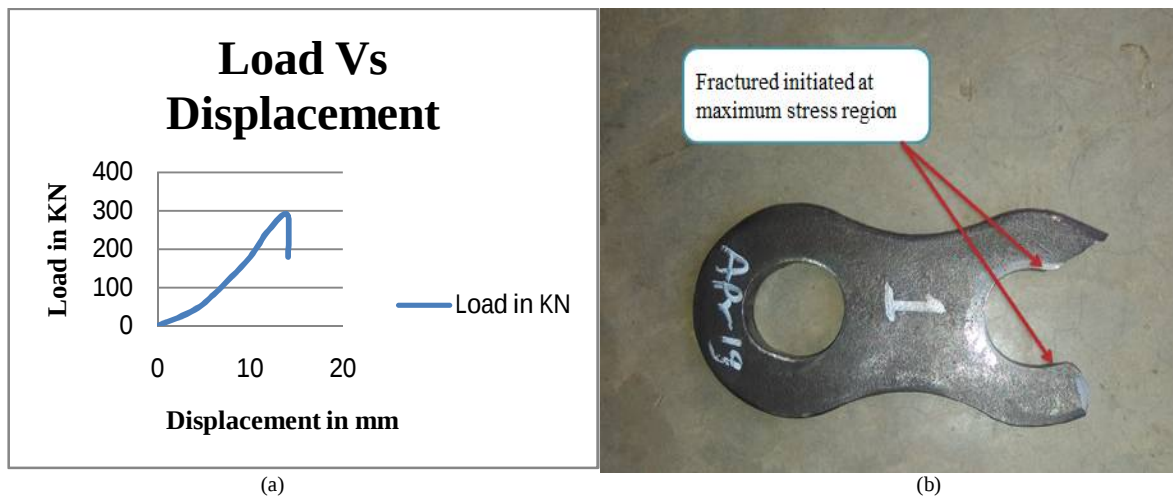


Fig.7. (a) Load versus displacement graph and (b) Broken chain link plate after tensile test

As the tensile load increases, failure occurs due to the fracture initiated at maximum stress region as shown in figure 7(b). It validates that maximum stress region in FEA and practical is same.

VII. RESULT AND DISCUSSION

Table II shows result of iteration carried out in shape optimization in ANSYS Workbench 15. The seventh iteration gives the lowest mass of the chain link plate for the required conditions, as stress is within 5 % of theoretical value, displacement is low.

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TABLE II
SHAPE OPTIMIZATION ITERATION

No. of Iterations	Maximum Von-Mises Stress in Mpa	Maximum Displacement in mm	Mass in kg
1	823.50	0.1085	0.504
2	815.99	0.1129	0.484
3	816.69	0.1109	0.475
4	813.49	0.1111	0.460
5	819.45	0.1155	0.445
6	822.53	0.1230	0.440
7	822.68	0.1122	0.431

Comparison of mass of chain link plate of conventional and redesigned is shown in table III

TABLE III
COMPARISON OF MASS BEFORE AND AFTER OPTIMIZATION

Mass of conventional link plate	Mass of redesigned link plate	Reduced mass in percentage
0.504 kg	0.431 kg	15 %

Comparison of theoretical and FEA Von-Mises stress is shown in table IV. Stress values are below the tensile strength of EN19 material. FEA Von-Mises stress is within 5% of the theoretical stress value. So we can say that design is safe.

TABLE IV
COMPARISON OF THEORETICAL AND FEA STRESS

Theoretical Stress	FEA Von-Mises Stress
821.45 Mpa	822.68 Mpa

Comparison of theoretical and experimental breaking load of redesigned chain link plate is shown in table V. Experimental breaking load is greater than that of the theoretical, so design is safe.

TABLE V
COMPARISON OF BREAKING LOAD

Theoretical Breaking load	Experimental Breaking load	Remark
269.77 KN	290 KN	Design is safe.

VIII. CONCLUSION

1. Weight of the link plate before optimization is 0.504 kg and after shape optimization weight is reduced to 0.431 kg. So, there is reduction in weight by 15 %.
2. It has been observed that there was high stress concentration near to hole in the chain link plate therefore breaking takes place at the location of minimum cross section area.
3. As the Finite element analysis results are within +/- 5% of the calculated working stress, so the chain link plate, pin and assembly were safe under the maximum working load of 267KN.
4. The experimental breaking load of EN19 link plate is 290 KN, which is greater than the working load value 267 KN. So, it shows that the optimized design is safe.

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