DESIGN OF CHAIN LINK FOR DRAG CHAIN CONVEYOR TO INCREASE BREAKING STRENGTH

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ABSTRACT

A conveyor system is a mechanical handling equipment that moves material from one location to another. Chain conveyors are designed to convey wide variety of powdery, granular & lumpy bulk materials. Forged chain links are used in drag chain conveyors in various industries. The inclination in demand rate requires high quantity of material to be transported in minimum amount of time. Due to this the current installed system has to be redesigned and considered for an increased tonnage carrying capacity, which in turn calls for stronger design of components. In this project, work has been done to increase the chain link breaking strength from 40T to 70T for higher models without changing the pitch, 216 mm. To achieve the target, critical sections of existing 216 mm pitch chain link, in terms of stress are found out and the design is modified in those critical sections. This is carried out in two options. First option is 'Modified dimensions with existing material'. And in second option, suitability of alternative materials is checked and new material is suggested without changing the dimensions. Best suited material is selected on account of suitability for manufacturing processes, availability, and costs involved. Finite Element Analysis with ANSYS14.0 is used to validate both the solutions. Option I and option II are compared in terms of advantages, disadvantages, manufacturing process, costs involved, time required to implement to find out the best suitable option.

Keyword: - *chain link, drag chain conveyor, breaking strength*

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1. INTRODUCTION

Chain links are used in conveyors to carry material from one point to other. Such conveyors are used in various sectors such as cement, power, fertilizer etc.

Since chains have been in use for lifting and fastening application for a long time, many topics related to conveyor chain links' failures, fatigue analysis in operation have been discussed in the past. It can be stated that life of chain link depends on chain link manufacturing processes, stresses developed during operation and loading cycle. However efforts have not been focused on one area i.e. the increase in breaking strength of the chain link used in drag chain conveyors. In this paper, efforts are concentrated to increase the breaking strength of the chain link used in drag chain conveyors. [1, 2]

A conveyor system is a mechanical material handling equipment that conveys material from one point to another. There are different types of conveyor systems based on the profile of conveyor, direction of material travel, carrying element etc. which include, chain conveyor, belt conveyor, deep pan conveyor, screw conveyor etc.

This paper is focused on alteration of chain link design in drag chain conveyor system. The main components of the equipment include chains, sprocket, drive mechanism and casings. [3] Numerous types of conveying equipments including drag chain conveyors are produced by Mahindra Tsubaki Conveyor Systems (MTC). A range of chain links are used in these conveyors based on chain pitch and breaking strength i.e. 100 mm pitch, 142 mm pitch, 216 mm pitch chains. MTC currently uses 216 mm pitch chain with breaking strength of 40 T. It has been observed by MTC that for high capacity of chain conveyors double strand chain is required to sustain high tensions developed in operation. [4]

1.1 Drag chain conveyor- chain tension calculations

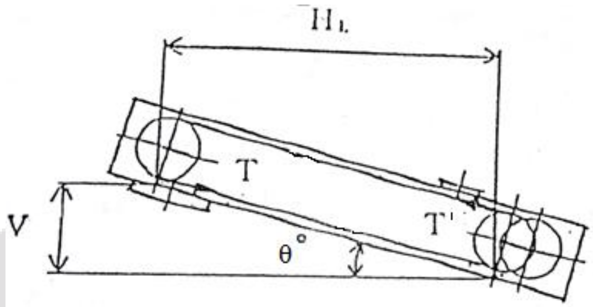

Figure-1: Schematic of Drag Chain Conveyor

Figure 1 shows the schematic arrangement of drag chain conveyor with terminologies indicated. [4]

Table -1: Terminologies used for chain tension calculation

Terminology	Description	Unit
Q	Capacity	TPH
M	Material weight	Kg/m
γ	Material Bulk Density	t/m3
Ƞv	Conveying efficiency	-
A	Cross sectional area	m3/m
S	Conveyor speed	m/min
HL	Horizontal C/C	m
V	Vertical lift	m
θ	Inclination	degrees
μF	Friction factor between material and casing	-
μN	friction factor between chain and casing	-
Cp	Chain Pitch	mm
W	Total chain + attachment weight	Kg/m
n	No. of sprocket teeth	-
T	Conveyor Chain Strength	kg

Table 1 describes the terminologies used in chain tension and capacity calculation with their units. [3]

Carrying Side Tension: T [kg]

$$T = (M \times \mu F + 2.1 \times W \times \mu N)\, HL + (M \times V)$$

Return Side Tension: T' [kg]

$$T' = W \times \mu N \times HL - (W \times V)$$

From above equations it can be seen that

T α M (rest are constants)

M= (1000 x Q) / (60 x S)

M α Q

Q = 60 x S x A x γ x Ƞv

All above equations implies

T α Q

High strength of chain link is required with increase in capacity of the conveyor. Hence requirement of this paper will be satisfied by increasing the breaking strength of the chain link which in turn will increase the conveying capacity of the drag chain conveyor.

2. PRESENT CHAIN LINK DESIGN

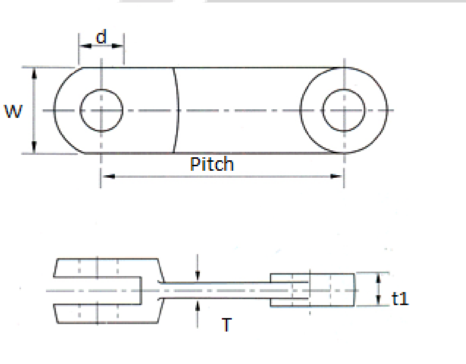

Figure-2: Schematic of Existing Chain Link

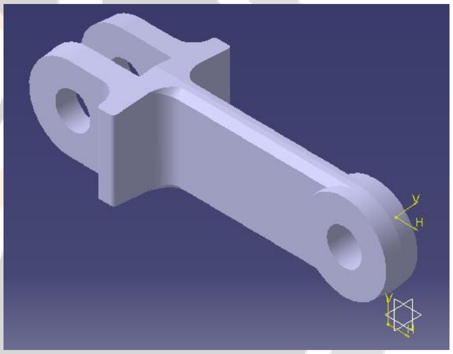

Figure-3: 3D Model of Existing Chain Link

Table-2: Present Chain Link Dimensions

Terminology	Description	Dimension (mm)
W	Diameter of fork	75
t_1	Thickness at single eyen end	62
t_2	Thickness of fork at double eye end	27
T	Thickness of link at web	18
d	Inner eye diameter	30

Drop forging process is used to manufacture the existing chain link. 20MnCr5 is the material of construction for the chain link. As 20MnCr5 can be heat treated to achieve desired surface hardness and required case depth for abrasive working conditions, it serves better in drag chain conveyor applications. Additionally it has good impact resisting properties, ductility and high wear resistance. Breaking strength of this chain link is 40 T. Mechanical Properties of the material are as given below [8]

Table-3: Mechanical Properties of 20MnCr5

Property	Unit	Value
Ultimate Tensile Strength after heat treatment	N/sq.mm	1000
Yield Strength after heat treatment	N/sq.mm	685
Young's Modulus	GPa	210
Poisson's Ratio	-	0.3

Break load test is carried out to check the breaking strength of chain links. Load is applied on assembly of two or three chain links held between one fixed and one movable jaw till chain link breaks. The set up for this process is as shown in below fig.

Figure-4: Set up for break load Test

3. IDENTIFICATION OF CRITICAL SECTION OF CHAIN LINK

To determine most critical section, stresses at various sections of existing chain link have been calculated. Chain link is considered to be under pure uniaxial tension. Chain dimensions are as mentioned in Table-3

Table-4: Input Data for chain link

Terminology	Unit	Value
Load Applied (P)	N	686700
FOS	-	1.2

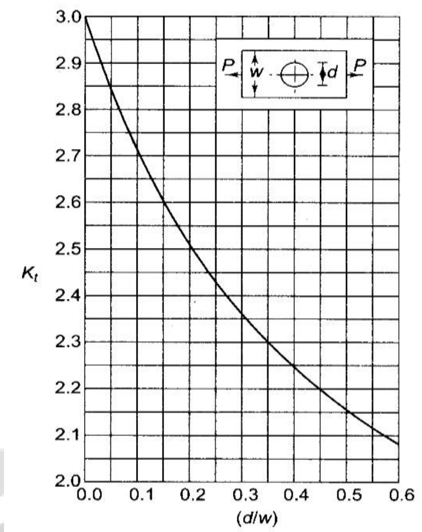

Figure-5.Stress Concentration factor K for a flat bar containing a circular hole [6, 7]

d/W = 30/75 = 0.4 ∴ K = 2.25 (refer **Figure-5**)

Ultimate Tensile Strength (Sut) = 1000 MPa

Allowable stress (σa) = Sut / FOS = 1000 / 1.2 = 833.33 MPa

The sections with stress more than allowable stress (σall) will be critical and hence to be modified.

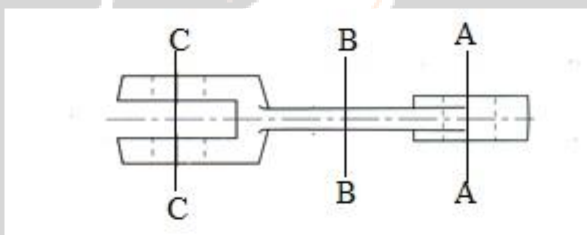

Figure-6 Sections of chain link

- Stress at section A-A (Refer **Figure-6)**

Nominal stress (σn) = P / [(W-d) x t1]

= 686700 / [(75-30) x 27] = 565.18 MPa

Maximum Stress at single eye end (σ1) = σn x K

= 565.18 x 2.2 = 1271.65 MPa. As, σ1 > σa , section A-A is critical

- Stress at section B-B (Refer **Figure-6)**

Stress at section B-B (σ2) = P / [T x W]

= 686700 / [18 x 75] = 508.67 MPa As, σ2 < σa , section B-B is not critical

- Stress at section C-C (Refer **Figure-6)**

Nominal stress (σn) = P / [(D-d) x t2 x 2]

= 686700 / [(75-30) x 17 x 2] = 448.82 MPa

Maximum Stress at double eye end (σ3) = σn x K

= 448.82 x 2.25 = 1009.84 MPa. As, σ3 > σa , section C-C is critical

Based on above calculations and available data of tests carried out on break load testing set up at MTC (refer **Figure-4**), it can be observed that section A-A & section C-C are critical. Hence, in this paper, efforts have been focused on to increase the breaking strength of these sections.

4. PROPOSED DESIGN OF CHAIN LINK

Alteration of 216 mm pitch chain link to increase its breaking strength has been carried out in following ways:

I. 216 mm pitch link is modified only with respect to dimensions and material is kept the same as 20MnCr5.
II. Options of substitute materials are explored and dimensions of the chain link are kept the same as existing.

4.1 Option I

It can be observed from calculation of stresses at critical sections of the chain link (refer point no **3)** that the strength of section A-A and section C-C (refer **Figure-6**) can be increased by increasing diameter of fork (W) and / or thickness of fork (t_1/t_2). Also it can be observed that as diameter of pin hole (d) reduces, stress concentration factor (k) increases which in turn increases the stress at section A-A and C-C, In addition to this reducing pin hole diameter will result into weakening the connecting pin between chain links. Hence scope of increasing strength by reducing pin-hole diameter (d) is very limited and hence same is excluded in this paper.

The factors for deciding optimum dimensions of modified chain link are stress at the section and weight of the chain link. Stress should be lower than the allowable stress. As cost of chain link is directly proportional to its price, weight should be as low as possible while satisfying the condition for stress. To find out optimum dimensions, stresses for a load of 686700 N are calculated at section A-A for range fork diameters while keeping all other dimensions constant. Also weight for each fork diameter (W) is calculated. For the same stress values thickness of fork at section A-A is calculated while keeping all the dimensions same as original chain link. Weight of for each Fork thickness (t_1) is calculated. The results are plotted on the graph of weight against stress as shown in fig below

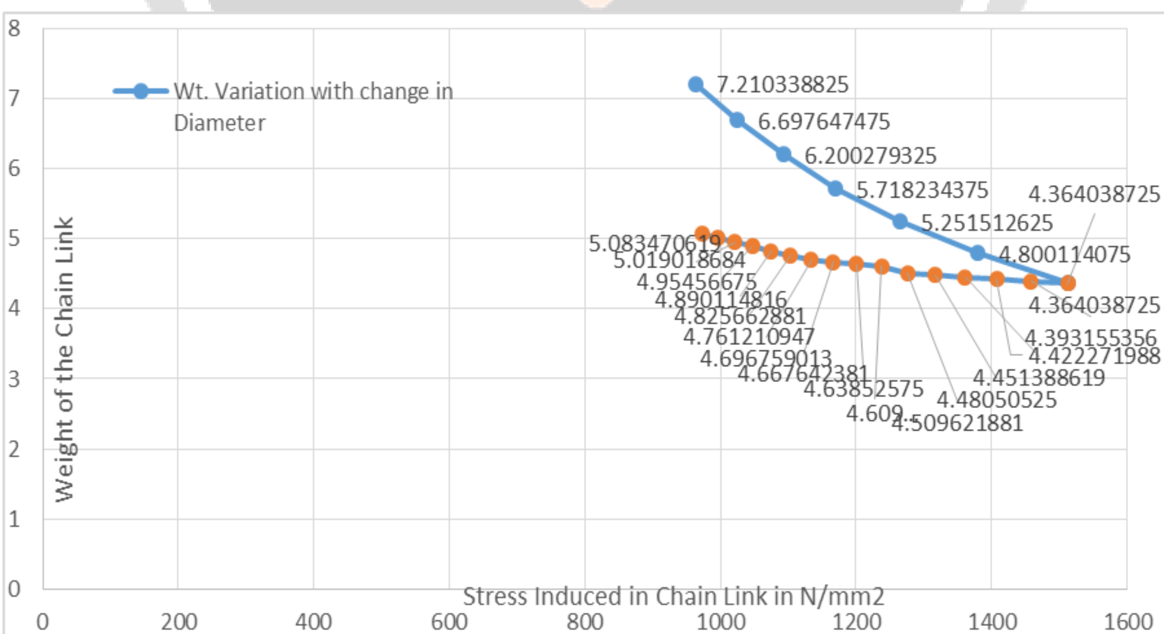

Chart – 1 Weight Vs Stress chart

It can be observed that, with increase in both dimensions stress reduces but at the same time weight increases. Slope of the graph for increase in respective dimensions suggests that up to a limit increasing fork diameter is advisable but past this limit weight of the chain link will increase significantly. Hence after such limit, satisfactory stress level for section A-A can be achieved by varying the fork thickness (t1). Similar analysis is carried out at section C-C. Thus following dimensions were selected on basis of this analysis

Table-5: New dimensions of chain link

Terminology	Unit	Value
Diameter of fork (W)	**mm**	**80**
Diameter of pin hole (d)	mm	30
Thickness of fork at single eye end (t1)	**mm**	**38**
Thickness of fork at double eye end (t2)	**mm**	**19**
Throat thickness (T)	mm	18

4.1.1 Validation -

d/W = 30/80 = 0.375 ∴ K = 2.27 (refer **Figure-5**)

Ultimate Tensile Strength (Sut) = 1000 MPa

Allowable stress (σa) = Sut / FOS = 1000 / 1.2 = 833.33 MPa

- Stress at section A-A (Refer **Figure-6)**

Nominal stress (σn) = P / [(W-d) x t1]

= 686700 / [(80-30) x 38] = 361.42 MPa

Maximum Stress at single eye end (σ1) = σn x K

= 361.42 x 2.27 = 820.4234 MPa.

As, σ1 < σa , section A-A is safe

The above design is also validated on ANSYS14.0

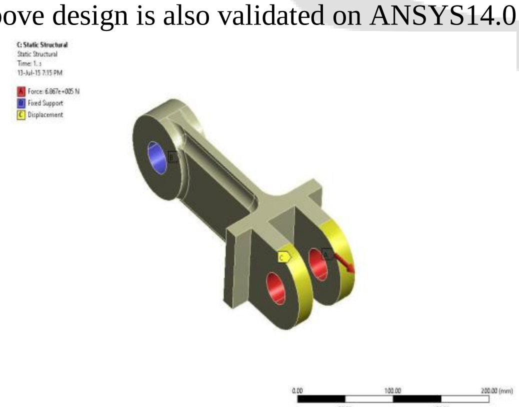

Figure-7.1 Applied boundary conditions

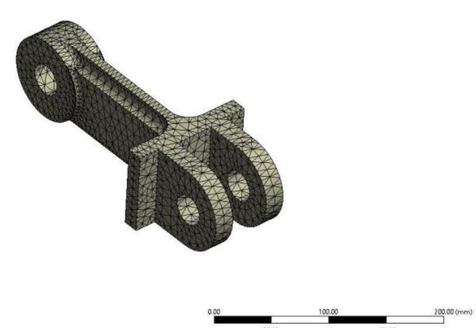

Figure-7.2 Meshing

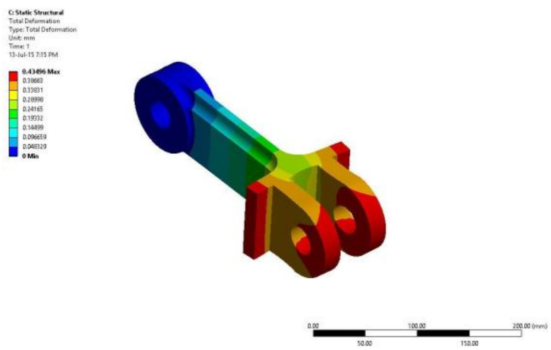

Figure-7.3 Deformation

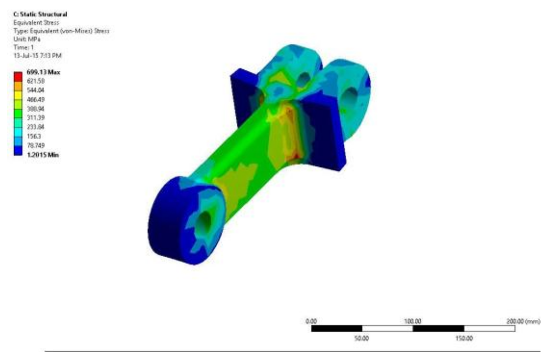

Figure-7.4 Maximum stress

Figure-7: FEA of proposed Chain link for Option I

Maximum Stress by Calculations = 820.42 MPa and Maximum Stress by ANSYS = 700 MPa. Maximum Stress by both methods is less than allowable stress that is 833.33 MPa. Hence our design is safe.

4.1.2 Costing

Cost per chain link is based on material cost, cost of forging, cost of heat treatment, and cost of machining. All these costs will be more than existing chain link cost as weight of new chain link is more. These costs are calculated per kg of the chain link weight. Weight of Present Chain link is 4.5 kg and Weight of Proposed Chain link is 5 kg. The additional cost to be taken into account for this option is cost of new die. Since, two dimension of the chain link have been changed, new die will be required for production of these chain links. Cost of new die will be one time investment.

4.2 Option II

It can be referred from calculation of stresses at critical sections (refer Point **– 3**) that the maximum stress for load of 686700 N is 1271.65 MPa and it appears at section A-A (refer **figure- 6**).

4.2.1 Calculation for desired S_{ut} for new material –

FOS = 1.2, Maximum Stress (F) = 1271.65 MPa

Desired S_{ut} = F x FOS = 1271.65 x 1.2 = 1525.9 MPa

Hence, desired S_{ut} for new material ≥ 1525.9 MPa

Materials meeting above requirement are as follows: [10]

Table-6: List of materials having ultimate tensile strength in range of 1525 -1550 [8]

Sr. no.	Material	Ultimate Tensile Strength (MPa)
1	30 Ni 16 Cr 5	1540
2	BS:970 EN30B	1540
3	40Ni10Cr3Mo6	1550
4	25 Cr 3 Mo 55	1540
5	40 Cr 3 Mo 1 V 20	1540
6	31 Ni 3 Cr 65 Mo 55	1540
7	40 Ni 3 Cr 65 Mo 55	1540
8	40 Ni 2 Cr 1 Mo 28	1540

4.2.2 Criteria for selection of material –

- Mechanical Properties

Materials enlisted in Table 4 are medium carbon steels. Carbon content in all these materials is between 0.25 to 0.70%. Plane carbon steel is especially compliant for machining or forging. In addition to this it can be heat treated to achieve suitable surface hardness.

Alloying elements-Nickel, Chromium, Molybdenum, and Vanadium etc impart following properties on above listed materials. [9]

- Hardness and machinability is improved by carbon.
- Refined grain structure and case hardening abilities are imparted by Nickel.
- Quenching speed can be reduced by virtue of Chromium which in turn improves wear resistance and toughness of steel.
- Ultimate strength of steel is increased without affecting ductility or workability by addition of Molybdenum in combination with chromium.
- Vanadium causes improve harden ability and increases resistance to softening at high temperatures.

Thus all the materials have suitable mechanical properties for the given application

- Suitability for Manufacturing processes

Sequence of manufacturing processes includes Forging, Machining and Heat treatment respectively. For forging material should have good ductility. For ease of machining material should have good mach inability and low hardness which can be increased to 52-56 HRC after machining processes by heat treatment. This hardness value is requirement of drag chain conveyor application.

- Availability

Availability of materials is checked across India.150 mm square bar is used as raw material.

4.2.3 Selected material – EN 30 B [9]

From list of materials (refer Table-5) EN 30 B is selected as most suitable material based on above criteria (refer point 4.2.2). Specifications for EN 30 B are as follows.

Chemical composition

Table-7 Chemical Composition of EN 30 B

Material	Percentage composition
Carbon	0.26-0.34%
Silicon	0.10-0.35%
Nickel	3.90-4.30%
Manganese	0.40-0.60%
Chromium	1.10-1.40%
Phosphorus	0.050% max
Molybdenum	0.20-0.40%
Sulfur	0.050% max

- **Form of supply –**
  It is supplied as round bar or plate – squares, flats and diameters can be sawn cut to required sizes. Ground steel bar can be supplied, providing a high quality steel precision ground bar to close tolerances. It is supplied in annealed condition so that it can be machined readily. The hardening treatment is relatively simple, quenching in air or oil from 810-830°C followed by tempering, with excellent mechanical properties being obtained.
- **Applications**
- EN30B alloy steel can be used for many purposes where toughness & high tensile strength are Requirements. For example: components of small presses including anvils, collars, strikers or hammers, rams or punch

holders. Other applications include rivet snaps, air hardening cold chisels, crimping tools, clutch keys, racks, pinions and angle pins for pressure die casting tooling. EN30B will machine readily in the annealed condition in which it is supplied.

For EN 30 B manufacturing processes and heat treatment procedures are as explained below

- **Forging**
  Steel is to be carefully heated to the forging temperature of 1000-1100°C, and soaked well. After forging, it is to be cooled slowly in a furnace to 100 Deg C maximum and annealed immediately.
- **Annealing**
  To obtain the softest condition the material is to be heated carefully to 630-650°C and soaked for a minimum of 2 hours, and then cooled in the furnace or in air. It is advisable to repeat this treatment to obtain the best machining characteristics.
- **Hardening**
  Material is to be heated uniformly to 810-830°C and when thoroughly soaked at this temperature, it is to be cooled in air or quenched in oil according to mass. Hardening from neutral salt baths will help to prevent scaling or decarburization and is strongly recommended. It is to be pre heated at 300-400°C, and raised to the hardening temperature of 810-830C and finally quenched into salt standing at 300-320°C. It is then to be withdrawn and cooled in air. Alternatively, components may be vacuum hardened. If desired, hardened and tempered components can be cyanide hardened to give a shallow carburized case to die surfaces with increased hardness values up to Rockwell 52-56 HRC.
- **Heat treatment**
  Heat treatment temperatures, including rate of heating, cooling and soaking times will vary due to factors such as the shape and size of each EN30B steel component. Other considerations during the heat treatment process include the type of furnace, quenching medium and work piece transfer facilities.

4.2.4. Validation

As EN 30 B has ultimate tensile strength as 1540 MPa, it is theoretically suitable for load of 686700 N. However same is validated with FEA on ANSYS 14.0

The Material Properties for EN 30 B are as follows.

For Design Case: - Material: Structural Linear Isotropic, Young's Modulus = 210000 N/mm2.

Poisson's Ratio = 0.3, Ultimate Tensile Strength: 1540 MPA, Yield Strength: 1240 MPA

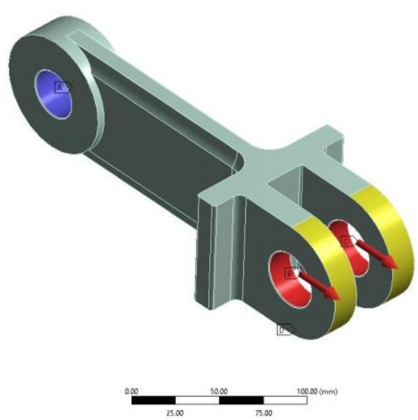

Figure 8.1: Boundary conditions

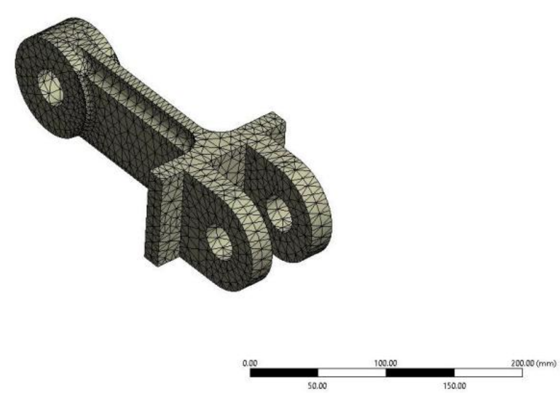

Figure 8.4: Meshing

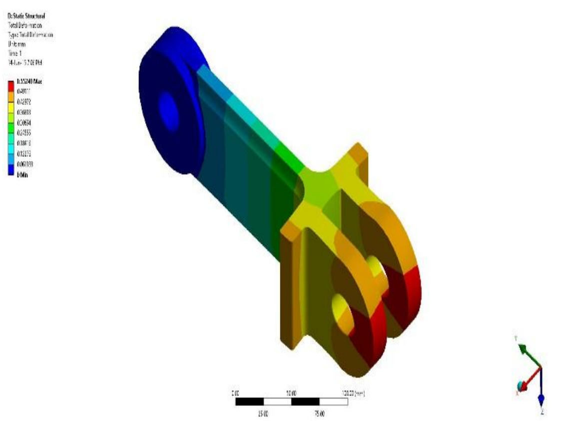

Figure 8.3: Deformation

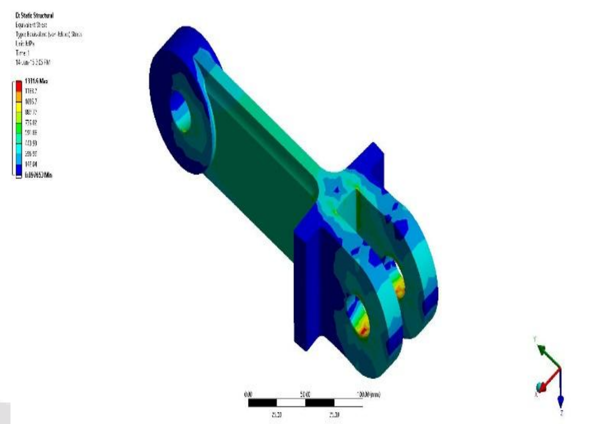

Figure 8.4: Maximum stress Calculations

Figure-8: FEA of proposed Chain link for Option II

Maximum Stress by Calculations = 1526 MPa and Maximum Stress by ANSYS = 1332 MPa. Maximum Stress by both methods is less than ultimate tensile strength that is 1540 MPa. Hence our design is safe.

4.2.5 Costing

Only material cost needs to be taken into consideration as dimensions of chain link are same,. Consequently there is no requirement of new die. Cost of forging, Cost of Heat Treatment, Cost of Machining all these costs will be same as that of present chain link. From quotations of various suppliers cost of EN 30 B is found to be greater than 20MnCr5.

5. RESULTS & DISCUSSIONS

To increase the breaking strength of the chain link, two ways are followed. In option I dimensions of the chain link are modified keeping the material of construction same as existing i.e. 20MnCr5. Modifications in the dimensions has been done by doing an exercise from which optimum dimensions that can be altered with optimum weight of the chain link. The dimensions modified are diameter of fork and Thickness of fork.

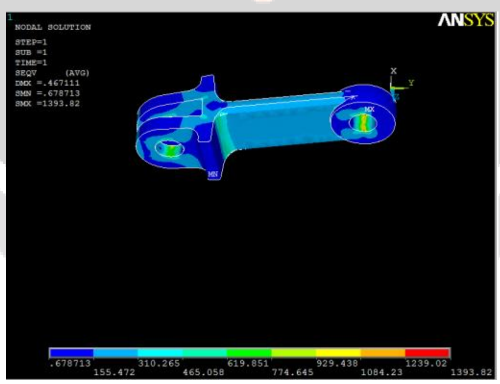

Figure 9: FEA of existing chain link after applying 70T load.

From the figure 9, max stress of 1393.82MPa on the 'I' end and min stress of 0.678713MPa on the fork end. The permissible tensile stress is 1000MPa. Hence this design is not safe for the given (70T) load.
However, objective is achieved and both new designs i.e. option I and option II are safe as shown in 4.1.1 and 4.2.4

Summary Table for option I:

Table 7.2 Summary Table for option I

Link	Pitch (P) mm	Fork Diameter (H) mm	Width of fork at double end (W2) mm	Width of fork at single end (W1) mm	Hole diameter (D) mm	Mass in kg
Existing link	216	75	62	27	30	6
Modified link	216	80	66	38	30	7

Validation for option I and Option II

Table 7.3 Summary Table of von mises stress for option I & option II

Approach	Breaking load in kg	Von-Mises Stress by Option I (MPa).	Von-Mises Stress by Option II (MPa).
Analytical method	70000	820	1526
FEA method	70000	700	1332

Ultimate tensile strength for 20MnCr5 is 1000 Mpa and for EN30B is 1540 MPa Von-Mises Stresses obtained by both methods is less than their ultimate tensile strength hence design is safe.

Comparison between option I and option II:

Table 7.5 Comparison between existing link, option I and option II

Sr. No.	**Parameter**	**Existing**	**Option I**	**Option II**
1	Breaking strength	40 T	70 T	70 T
2	Chain Pitch	216 mm	216 mm	216 mm
3	Dimensions	Existing	New	Existing
4	Fork Dia.	75	80	75
5	Thickness	27	38	27
6	Material	20MnCr5	20MnCr5	EN30B
7	Max. stress at 70T	1394 MPa	700 MPa	1332 MPa
8	UTS	1000 MPa	1000 MPa	1540 MPa
9	Design at 70 T	Fails	Safe	Safe
10	Raw material cost	X Rs./kg	X Rs./kg	3X Rs./kg
11	weight of link	4.5 kg	5 kg	4.5 kg
12	Cost of link	Y Rs.	1.5Y Rs.	1.8Y Rs.
13	Forging die	Existing	New Required	Existing
14	Forging die cost	NA	8 lakh Rs. Approx.	NA
15	Additional Cost	NA	25000 Approx.	NA
16	No. of strand required for 70T	2	1	1
17	Total Cost	2Y Rs.	1.5Y+8lakh+25k Rs.	1.8Y Rs.

From the above table, it can be seen that option II is viable solution as it has more advantages and quickly implementable. All other processes such manufacturing drawings, existing design of conveyor are same and need not to be changed hence it is ready to apply solution.

6. CONCLUSION

To increase the breaking strength of chain link, two options are applied, resolved and validated and best suited option is selected for implementation. Objective to increase the chain link breaking strength from 40T to 70T without changing the pitch, 216 mm. has been achieved by following two options.

- Option I: Modified dimensions with same (existing) material.
- Option II: Alternative material with same (existing) dimensions.

Option I has less advantages compared with option II. Option II is ready to implement. Also, there are no alterations in the existing design of chain hence can be immediately executed. In option I biggest disadvantage is new die requirement but in option II, by just changing the material from 20MnCr5 to EN30B, target is achieved. Increase in breaking strength from 40T to 70T ultimately increases the tonnage carrying capacity of the conveyor. Hence Option II is best suitable option for the industry.

Based on above discussion, it can be resolved that with increase in fork diameter and thickness of chain link, strength requirements have been met. However, it involves increased weight and ultimately cost. On the other hand, only changing the chain link material while meeting the strength requirements presents a viable solution from cost perspective as new die cost is totally eliminated in option II.

ACKNOWLEDGEMENT

The authors are thankful to the Mahindra Tsubaki Conveyor Systems Pvt. Ltd, Pune, India for giving the opportunity to work on this project.

REFERENCES

[1] Xiaolun Liu, Wencheng Wang, Wei Sun, Tiansong Wub,Jiajun Liua, Jianfang Liub, “Design and Experimental Analyse of Low Noise Double-Pitch Silent Chain for Conveyor” in 2012 International Workshop on Information and Electronics Engineering (IWIEE)

[2] Zheng Zhi-feng,Wang Yi-xing,Chai Bang-heng.Chain drive[M]. Beijing:Chain Machine Press, 1984

[3] MENG Fan-zhong. The meshing principle of silent chain[M].Beijing:Chain Machine Press, 2008

[4] Khaled Al-Fadhalah , Ahmed Elkholy, Majed Majeed, “Failure analysis of Grade-80 alloy steel towing chain links”, ELSEVIER Engineering Failure Analysis 17 (2010) 1542–1550

[5] M. Sujata, M.A. Venkataswamy, M.A. Parameswara, S.K. Bhaumik, “Failure analysis of conveyor chain links”, ELSEVIER-Engineering Failure Analysis 13 (2006) 914–924

[6] Ishwar G. Mulani, “Engineering Science And Application Design For Belt Conveyors” [2012]

[7] Complete Guide to Chains, Tsubakimoto Chain Company. [2000]

[8] MTCS catalogues, standards – Mahindra tsubaki conveyor systems Pvt. Ltd. (2000)

[9] MTCS MAT Standards, Supply specification No. MECP/ENGG/MAT/004 – Rev 0 (2000)

[10] V B Bhandari, “ Design of machine elements” Third Edition, Tata McGraw-Hill Education, 2010

[11] S. Ramamrutham,, R. Narayanan, “Strength of materials” 14th edition , Dhanpat Rai Publishing company.

[12] Design Data Book, Faculty of Mechanical Engineering, PSG College of Technology, Coimbatore, PG 1.18

[13] Séblin. B, Jahazeeah. Y, Sujeebun. S, Manohar, Wong Ky. B Booklet on Material Science- MECH 2004.