International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395 -0056
Volume: 03 Issue: 02 | Feb-2016 www.irjet.net p-ISSN: 2395-0072

"Design Optimization of Roller Chain Link Plate used in Sugar Industry"

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Abstract - *Chain Link assembly is extensively used in the sugar industry. It was determined that maximum amount of weight of chain conveyor is covered by the roller chain link plate Finite Element Analysis (FEA) has used to conduct shape optimization and weight optimization of roller chain outer link plate. We have concentrated on outer link plate and weight reduction of link plate by changing the shape of outer link plate. The aim of this paper is design optimization of roller chain link plate. In this paper the analytical and numerical methods are used for optimization of roller chain link plate. Also the experimentation has done to check validation of the work. The advantage of this paper is that it saves 72gm of weight per link plate and 1.2 kg per meter length of chain.*

Key Words: *Roller chain, Link plate, FEA analysis, design optimization, UTM.*

1.INTRODUCTION

Failure of chain is perennial problem in industries which causes huge losses to these industries along with its dependents and in turn economical growth of the state. So, roller chain is the most important element of the industrial processes.

Fig. 1.1: chain link assembly at sugar industry.

This paper contain FEA Based Design and experimental validation of outer Chain Link plate used in Roller Chain because that area is one of the most weighted. Besides, this paper will see the system in terms of chain link plate optimization and operation for improvements.

2. LITERATURE REVIEW

Chain Design optimization has been a hot area of research where several patents are filed. The key paper and patents filed regarding chain design are discussed here.

Payet P.R, " Process and Conveyer Device for Feeding Sugar Cane in a Mill Train", U. S. Patent # 3, 120, 173, 1964. [1]Initial work on conveyer device includes improving the sugar cane conveying process at required speed in a mill train. The purpose of the invention was to produce a device enabling very high conveying speeds.

White D.C. and Fraboni H. J. "Roller Chain Sprockets Oriented to Minimize Strand Length Variation", United States Patent # 6213905, 2001[2] suggested an arrangement of a roller chain and sprocket drive which utilizes sprockets having randomized radial seating positions, with the sprockets arranged within the drive to be in an orientation with each other to provide favorable dynamics to the roller chain and sprocket drive. This also helps in minimization of strand variation which does exist out of phase for sprockets.

Some patents related to roller chain conveyors are also filed for improving its performance and endurance. Hodlewsky W.G. ,"Conveyor Chain Assembly", U.S. Patent # 4,858,753, 1989.[3] designed a conveyor chain in which hinge pin joining the links can be inserted and held in place without the necessity of forming a head on the pin thereby reducing the labor and cost required in the assembly of the chain links.

Seymour T. H. "Chain and Sprocket Combination", U.S. Patent # 5,127,884, 1992.[4] suggested chain and sprocket combination that reduces wear and allows for an increasing in the speed of operations of the devices that use chain and sprockets.

From the earlier studies, it can be illustrious that, even though some patents are filed on roller chains most of the patents are built on improvement of efficiency and performance. Hardly limited patents are there on improving life of the chain and minimization of its failure and weight. It can also be noted that the analytical work in the literature is focused on load estimate. Very rare researchers have explored the stress analysis for the chain assembly and chain link plate. From the chain failure case studies it can be noted that the root cause of failure was fault, design and weight of chain assembly. After survey from industry and collecting the problems we can conclude that high weight of chain is acting as a major problem in addition to all fixed problems like excess loading. Excessive wear of the chain link plate, wear of sprocket due to weighted chain is also the main problem and it happens due to sliding of weighted roller chain link plate and sprocket as industry is running for 24hr. for 6 to 8 months hence there is motivation to reduce the weight of chain link assembly. Slippage of chain due to misalignment of sprockets and vibration of chain due to high weight of chain itself gives motivation to reduce the weight of chain. More power consumption to rotate drive shaft as it has already 12 tone load, weighted roller chain adds unnecessary weight and again gives motivation to reduce the weight of chain, so there is motivation for design optimization of chain link assembly, but as we consider the chain link assembly most of the dimensions are fixed and all are in proportion with the pitch of the chain. The weight saving area in chain link assembly is only in roller chain link plate because it has a high weight as compare to other component of chain link assembly like pin, bush and roller. As other components are also related to design of sprocket so we can't change other parts dimensions. Pitch of the chain is also constant so we have to keep pitch of the chain constant while optimization of roller chain link plate. So we have motivation to optimize the design of roller chain link plate to reduce the weight of it keeping the strength constant for a given loading conditions.

3. DESIGN PROCESS BASED ON BREAKING LOAD

3.1 Calculations for outer Chain link plate[10]:

As per the catalogue we had taken chain link plate of dimensions 68.30mm x 76.20mm (Pitch) x 9.5mm for outer link plate and 68.30mm x 76.20mm (Pitch) x 11.30mm for inner link plate. EN-19 material. Now we can find out the value of ultimate tensile strength i.e. maximum stress. by using the analytical formulae

Tensile strength = 1097 N/mm^2 to 1230 N/mm^2

Poisson's ratio = 0.3

Modulus of elasticity = 2.05X10^5 N/mm^2

Maximum working stress-

$$MaximumWorkingstress = \frac{Maximumstress}{Factorofsafety}$$

$$MaximumWorkingstress = \frac{1230}{1.5}$$

$$MaximumWorkingstress = 820\ N/mm2$$

Minimum working stress-

$$MinimumWorkingstress = \frac{Minimumstress}{Factorofsafety}$$

$$MinimumWorkingstress = \frac{1097}{1.5}$$

$$MinimumWorkingstress = 731.33\ N/mm2$$

So as per the above analytical calculations we got maximum working stress of 820 N/mm^2 and minimum working stress of 731.33 N/mm^2. Now by using that working stress values and by using the following formulae we can calculate the working load that the chain link plate can carry.

Maximum working load for outer chain link plate-

$$MaximumWorkingstress = \frac{WorkingLoad}{ResistingArea}$$

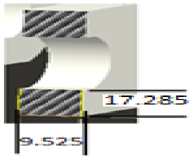


Fig. 2 Resisting area of link plate

$$820 = \frac{Maximumworkingload}{2\ (17.285\ \times 9.5)}$$

$$MaximumWorkingLoad = 820 \times 328.415$$

$$MaximumWorkingLoad = 269{,}300.03\ N$$

Minimum working load for outer chain link plate-

$$Workingstress = \frac{MinimumWorkingLoad}{ResistingArea}$$

$$731.33 = \frac{Minimumworkingload}{2\ (9.5 \times 17.285)}$$

$$MinimumWorkingLoad = 731.33 \times 328.415$$

$$MinimumWorkingLoad = 240336.98\ N$$

From above calculation we got the working load range i.e. varies from **2,69,300.3**N to 2**,40,336.98**N for chain outer link plate. So as per the above analytical calculations we got the maximum working stress of 820 N/mm^2 and minimum working stress of 731.33 N/mm^2

The major dimensions of standard roller chains are approximately proportional to the chain pitch. These proportions were derived from detailed engineering studies and much experience. These proportions give an excellent balance of properties needed for a roller chain to perform well in a wide variety of applications. The approximate proportions of standard roller chains, conforming to ASME B29.1, are listed below[19]

- Chain (roller) width ≈5/8 of the pitch.
- Roller width ≈ 5/8 of the pitch.
- Link plate thickness ≈ 1/8 of the pitch.
- Pin diameter ≈ 5/16 of the pitch.
- Maximum pin link plate height ≈ 0.82 × pitch.
- Maximum roller link plate height ≈ 0.95 × pitch.

Old design:

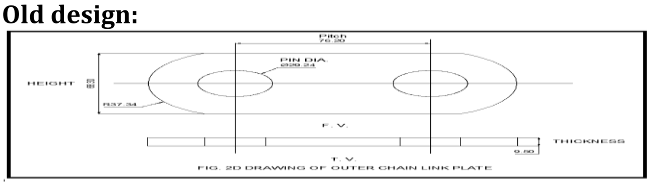


Fig3. Existing typical link plate dimensions

New design concept:

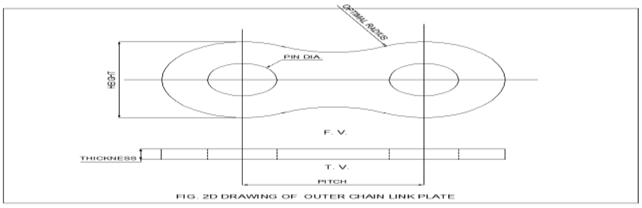


Fig 4 New design concept as per ASME B29.1 standard [16]

We can design the existing roller chain link plate with new ASME standard keeping the pitch and strength constant.

Table 3.1 Material properties for Link plate

Sr. No.	Parameter	Descriptions
1	Material	EN19
2	Young's Modulus E	2.05x105 N/mm2
3	Density ρ	8.00X10-9 Tons/mm3
4	Poisson's ration	0.3
5	Tensile Strength	1230N/mm2

As per ASME Standard baseline design model is considered as a Concept-1. Within the outer link, most dimensions in the industry are parametrically defined, however one dimension, the radius that is in between the inter connecting holes is left to manufacturer convenience. So in Concept-1 fully rectangular design is considered of a given size 144.5mm*68.3mm*9.5mm,

Concept 1:

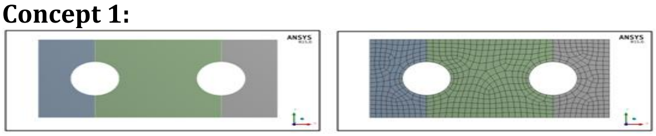


Fig. 5 Geometry and Mesh model of Concept-1

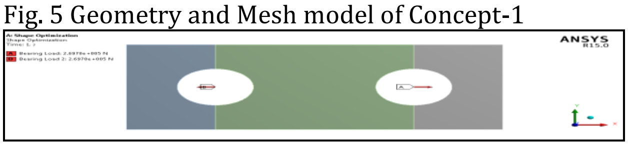


Fig. 6 Boundary Conditions on Concept-1

Above Fig. 5 shows the Geometry and Mesh model of Concept-1. And Fig. 6 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 as shown in the above.

Fig. 7 Shape Finder plotConcept-1

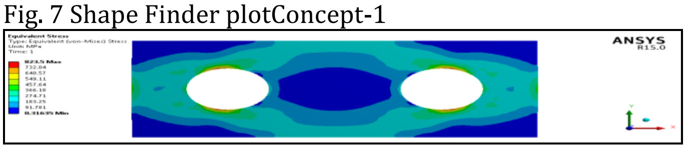


Fig. 8 Von Misses Stress plot of Concept-1

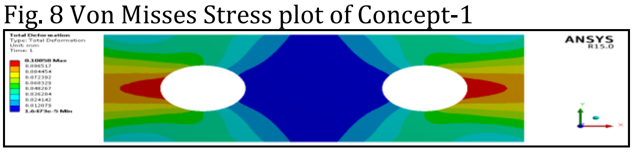


Fig. 9 Displacement plot of Concept-1

Above Fig. 4.9 shows Displacement plot of Concept-1 after running the static analysis in shape optimization module in ANSYS Work Bench. From Von Misses Stress plot as shown in Fig. 8 and Shape Finder plot as shown in Fig. 4.6, 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.

Concept 2:

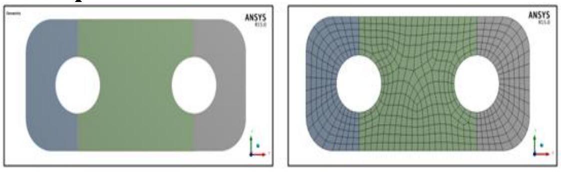


Fig. 10 Geometry and Mesh model of Concept-2

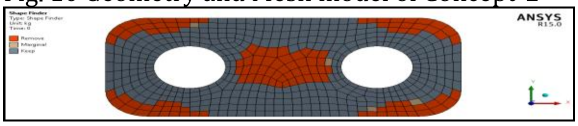


Fig. 11 Shape Finder plotConcept-2

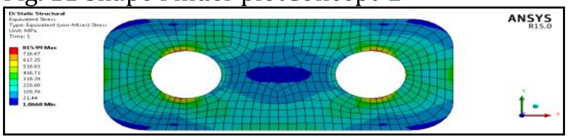


Fig. 12 Von Misses Stress plot of Concept-2

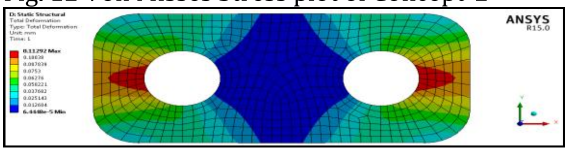


Fig. 13 Displacement plot of Concept-2

From Equivalent Stress plot and Shape Finder plots of this iteration as shown in Fig 12 and Fig. 11 respectively it is concluded that more material can be removed from the edge area as stresses observed are still very low and well within the acceptance limit. It comes to the next iteration again by following the contours of stress plots and shape finder.

Fig.13 shows Displacement plot of Concept-2 after running the static analysis in shape optimization module in ANSYS Work Bench. From Von Misses Stress plot as shown in Fig. 12 and Shape Finder plot as shown in Fig. 11, the next iteration of out model arises. Corners of the plates are removed from link plate denoting the contours of the low stress regions and shape finder plots. Link plate used is doomed to be used for load transmitter it needs to be rigid and require additional rigidity at its peak load at which high stress attention near hole locality may occur. Thus, after seven repetitions ANSYS solver provides an optimal shape as shown below.

Concept 7:

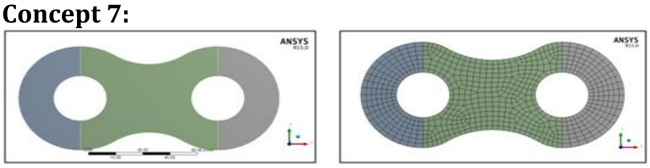


Fig. 14 Geometry and Mesh model of Concept-7

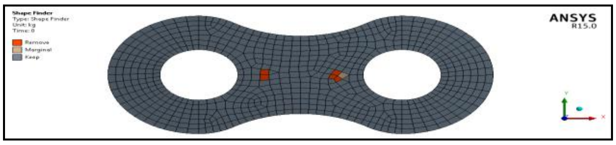


Fig. 15 Shape Finder plot Concept-7

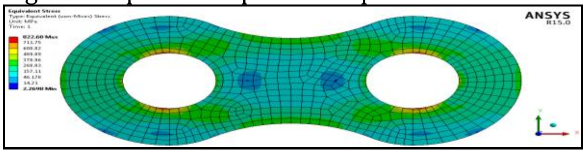


Fig. 16 Von Misses stress plot of Concept-7

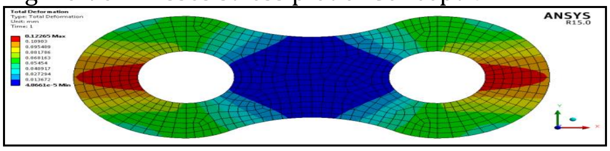


Fig. 17 Displacement plot of Concept-7

Final iteration above shows evolved model through history followed by engineering statistics to gain required link model as per standard topology optimization process.

3.2 Design Optimization of Outer Link Plate:

The chain link plate is first modeled in CATIA V-5 as shown in Fig. 18 and then exported to ANSYS where it is further meshed, constrained and loaded and simulated further.

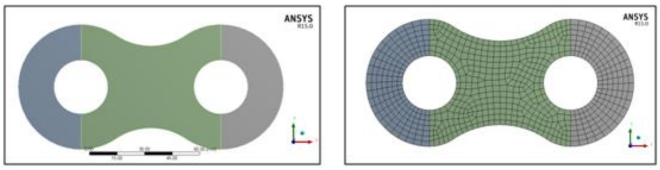


Fig.18 Geometry with Meshed model of Outer Link Plate

Meshed model of chain link plate is also shown in Fig. 18. The chain link plate has been meshed by using rectangular grid. A material property for EN-19 has been entered as an input. For meshing 5mm element edge length is considered 2,884 elements and 14,922 nodes are generated to mesh the chain link freely.

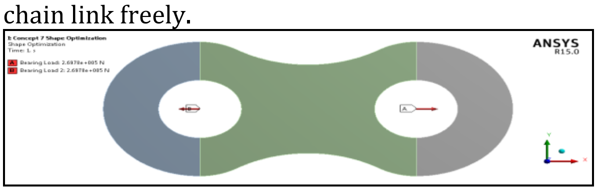


Fig. 19 Boundary Conditions for Outer Link Plate

In the DOE study, however, the sampling points increase dramatically as number of input parameters increases. For example, a total of 149 sampling points (finite element evaluations) are needed for 10 input variables using Central Composite Design with fractional factorial scheme. As the number of input variables increases, the analysis becomes more and more intractable. In this case, one would like to exclude unimportant input parameters from the DOE sampling in order to reduce unnecessary sampling points.

Table 3.2 Design of Experiment for Outer Link Plate

Table of Schematic B2: Design of Experiments (Custom + Sampling : Total Number of Samples = 15)

	A	B	C	D	E	F	G
1	Name	P1 - Height (mm)	P2 - Rad (mm)	P3 - Thickness (mm)	P4 - Equivalent Stress Maximum (MPa)	P5 - Total Deformation Maximum (mm)	P6 - Geometry Mass (kg)
2	8	61.47	40.5	8.55	1083.4	0.17029	0.31981
3	10	61.47	49.5	8.55	1020.4	0.15109	0.3382
4	2	61.47	45	9.5	947.31	0.14435	0.365
5	16	64.78	40.557	9.4906	898.74	0.1432	0.38748
6	6	68.3	45	8.55	913.43	0.13924	0.3888
7	12	61.47	40.5	10.45	886.04	0.13965	0.39088
8	14	61.47	49.5	10.45	834.13	0.12393	0.41335
9	1 (DP 0)	68.3	45	9.5	821.87	0.12546	0.432
10	5	68.3	49.5	9.5	800.97	0.11905	0.4404
11	9	75.13	40.5	8.55	833.99	0.13123	0.45353
12	11	75.13	49.5	8.55	799.85	0.11911	0.46308
13	7	68.3	45	10.45	747.16	0.11418	0.4752
14	3	75.13	45	9.5	735.72	0.11254	0.5087
15	13	75.13	40.5	10.45	682.46	0.1076	0.55432
16	15	75.13	49.5	10.45	654.32	0.09768	0.56599

i. Von Misses Stress Plot for outer chain link:

As shown in Fig. 20 red color 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, 69,300N it can sustain maximum stress of 820N/mm2

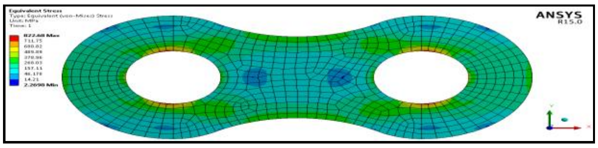


Fig 20 Von Misses stress plot for outer link plate

ii. Displacement Plot for Outer link plate:

Fig. 21 shows distribution of displacement along the length of chain link plate. It is observed that there is maximum displacement at the minimum cross sectional area along the length. Its value is 0.1226 mm and displacement becomes zero at the middle. So there are more chances of failure near to hole at the red colored region.

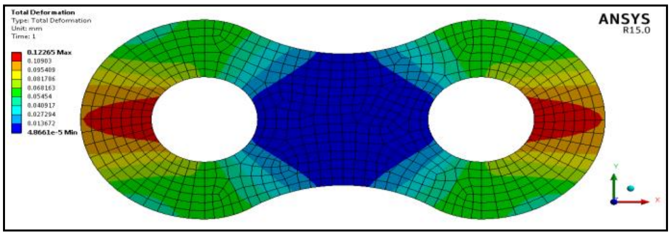


Fig. 21 Displacement plot for outer link plate

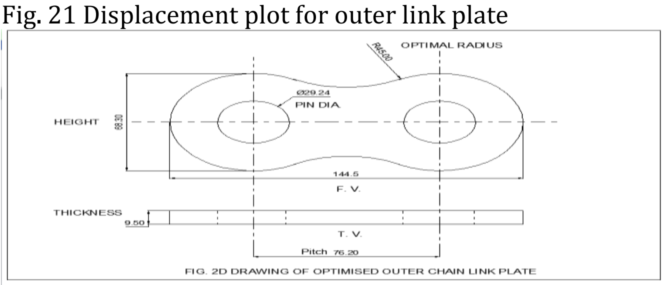


Fig. 22: 2D drawing of optimized link plate

4 EXPERIMENTAL WORKS

4.1 Testing of chain links on UTM:

Experimental testing of chain link plates is carried out to study the effect of material, for this testing we have used a Universal Testing Machine of 100 Tons capacity.

Results and Discussion

Table 4.1.Result of tensile test conducted on four optimized outer link plates

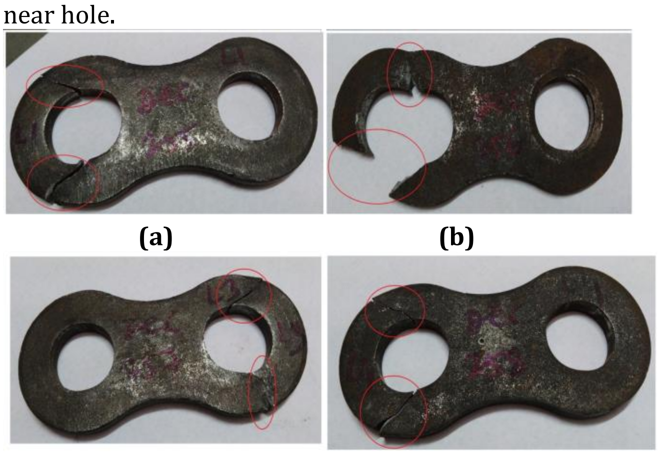
Link Plate	Breaking Load (N)	Max. Displacement(mm)
L1	28,1800	17.700
L2	27,1350	15.95
L3	29,7650	13.80
L4	28,1050	15.50

4.2 Behavior of single link plate under loading conditions:

Experimental testing of chain link plate of EN-19 material has been performed on Universal Testing Machine. The dimension of optimized Outer link plate was 68.30 mm x 76.20 mm (Pitch) x 9.5 mm

Chart. Graph of displacement v/s load for a single link plateL1 (Outer).

The above chart shows the graph of displacement v/s load for a single link plateL1 (Outer). Displacement is taken on X-axis and load in kN is taken on Y-axis. The max load of 281.800kN is taken by the chain link plate and then it fails near hole.



(a) (b) (c) (d)

Fig.23 Failure location of Outer Link Plate

The above Fig. 23 (a), (b), (c)& (d) shows the Outer link plate fails near to the hole. It shows that failure takes place at minimum cross section area of outer link plates.

4.3 Comparison of Theoretical, Finite Element Analysis and Experimental Test Results:

Theoretical calculation of allowable maximum stress, Finite element analysis and experimental work has been carried out for Outer link plate. Below table gives the comparison results for theoretical, numerical (Finite Element Analysis) and Experimental test for outer link plate under the static tensile loading condition.

Table 4.2 Comparison of Test Results of Chain link plate

Sr. No.	Theoretical Stress (N/mm^2)	Von Misses Stress (N/mm^2)	Theoretical Breaking Load (N)	Experimental Breaking Load (N)
L1	820.00	821.87	269300	28,1800
L2	820.00	821.87	269300	27,1350
L3	820.00	821.87	269300	29,7650
L4	820.00	821.87	269300	28,1050

The above Table 4.2 shows theoretical stress of outer link plate is 820.00$N/mm^2$and as per numerical it is coming 821.87 N/mm^2 and it is less than allowable limit, so the chain link plate design is safe. Also its breaking load is 2,82,962.5 N (28 Tons).

4.4 Weight optimization Results of link plates:

On the basis of Theoretical, FEA and Experimental results, it is clearly indicate that the optimal value of radius is 44 to 45mm, thickness is 9.5mm and height of the link plate 63.8 mm is observed. However this optimization seems like insignificant on its individual, it must be well-known that in a typical industrial application, thousands of such links will be needed. The weight saving thus achieved (Typical chain link plate 504 gm and chain link optimized 432 gm) 72 gm per link plate and 1.5 Kg per meter of roller chain will have a major impact on cost of the chain, and with a lighter chain, the cost savings during operation will also be significant.

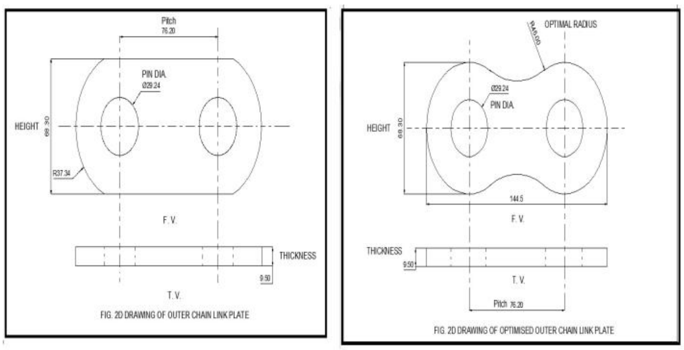


Fig. 24 Roller chain link plate before of link plates before optimization and after optimization

5 CONCLUSIONS

Based on the FEA and Experimental results, it is observed that the optimal value of radius is between 44.5 to 45 mm, thickness is 9.5mm and height of link plate is 68.30mm is observed. The weight saving thus achieved (Typical chain link 504 gm and optimized chain link 432 gm) 72 gm per link plate and 1.5 Kg per meter of roller chain have a major impact on cost of the chain, and more importantly with a light weight chain, the cost savings during operation is also significant. We studied the analytical, experimental and numerical behavior of link plates under tensile loading. As the Finite element analysis results are within +/- 10% of the calculated working stress, so the chain link plate were safe under the maximum working load of 26 Tons. The weight saving achieved have a significant impact on reducing wear of sprocket and roller chain link plate itself it also impact on reduction in power consumption of motor as chain weight reduced, it also effects on reducing the centrifugal force and unnecessary movement because of it.

REFERENCES

[1] Payet P.R," Process and Conveyer Device for Feeding Sugar Cane in a Mill Train", U. S. Patent # 3, 120, 173, 1964.
[2] White D.C. and FraboniH.J.,"Roller Chain Sprockets Oriented to Minimize Strand Length Variation", United States Patent # 6213905, 2001.
[3] HodlewskyW.G.,"Conveyor Chain Assembly", U.S. Patent # 4,858,753, 1989.
[4] Seymour T. H., "Chain and Sprocket Combination", U.S. Patent # 5,127,884, 1992.
[5] Garbagnati C., "Chain Conveyor Link Provided With a Pin-Locking Device, U.S. Patent # 5,293,989, 1994.
[6] Ensch P. J., Marks P. J., Montaine T. and Schladweiler, T. J., "Direction Changing Mechanism For Transferring Articles Between Transverse Conveyors", U.S. Patent # 5,634,550, 1997.
[7] Shibayama K. and Murakami Y., "Slat Conveyor Chain", U.S. Patent # 6,189,686 B1, 2001.
[8] Marsetti S. and Realtti R., "Chain Conveyor", U.S. Patent # 6,763,936 B2, 2004.
[9] Low K. H., "Computer-Aided Selection of Roller Chain Drives", Computer and Structures, 55, 925-936, 1995.
[10] M. D. Jagtap, B. D. Gaikwad, P. M. Pawar, B. P. Ronge(2014), "Use Of Strain Gages To Analyze The Behavior Of roller Conveyor Chain Strip," International Conference on Computer Science and Mechanical Engineering, 10th August 2014, Jaipur, India, ISBN: 978-93-84209-42-1.
[11] NurIsmalinaBintiHaris, "Failure Analysis of Conveyor Chain Linksa Case Study at Top Glove Sdn. Bhd. July 2013
[12] Sadagopan P., Rudramurthy R. and Krishnamurthy R., "Wear and Fatigue Analysis of Two Wheeler Transmission Chain", Journal of Scientific and Industrial Research, 66, 912-918, 2007.
[13] Noguchi S., Nagasaki K., Nakayama S., Kanda T., Nishino T. and Ohtani T., "Static Stress Analysis of Link Plate Of Roller Chain using Finite Element Method and Some Design Proposals for Weight Saving," Journal of Advanced Mechanical Design, Systems, and Manufacturing, No.09-0122. 2007.
[14] James A., "Failures of Chain Systems", Engineering Failure Analysis, 4(1), 57-70, 1997.
[15] Sujata M. Venkataswamy M. A., Parameswara M. A., and Bhaumik S. K., "Failure Analysis of Conveyor Chain Links", Engineering Failure Analysis, 13(6), 914-924, 2006.
[16] American Chain Association, Second Edition, "Standard Hand Book of Chains- Chains for Power Transmission and Material Handling," 2006.
[17] Tushar D. Bhoite, Prashant M. Pawar & Bhaskar D. Gaikwad "Fea Based Study of Effect of Radial Variation of Outer Link in A Typical Roller Chain Link Assembly" (2012), International Journal of Mechanical and Industrial Engineering (IJMIE), ISSN No. 2231 –6477, Vol-1, Issue-4, 2012
[18] PSG College of Technology, Design Data Book of Engineers, Kalaikathir
Achchagam, April 2007.
[19] ASME B29.1M, An American National Standard for precision power transmission roller chains, Attachments, and Spockets, Revision of ASME/ANSI B29.1M-1986, 345 East 47th street, New York, N. Y. 10017