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“DESIGN IMPROVEMENT AND FINITE ELEMENT EVALUATION OF A 102 MM PITCH DRAG CONVEYOR CHAIN: LINK THICKNESS SELECTION, CHAIN PULL, AND FASTENER REDESIGN”

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ABSTRACT

This study presents the design, finite element evaluation, and selection of a 102 mm pitch drag conveyor chain for flat-bottom drag conveyors that handle grain in troughs of 250 mm and 300 mm width and lengths of 100–150 m. The chain components were modelled in SolidWorks following the Indian standard IS 6834 and the Japan Chain Industry Association standard, and the inner and outer link plates of 5, 6, and 8 mm thickness were analysed by linear static finite element analysis in HyperMesh and OptiStruct under tensile loads of 10–50 kN. At a load of 10 kN, the maximum von Mises stress in the outer link decreased from 146.8 MPa for the 5 mm plate to 122.2 MPa for the 6 mm plate and 91.56 MPa for the 8 mm plate, and the inner link carried about 28–31% higher stress than the outer link. The chain pull was calculated with the Renold relation and increased from about 2.8–3.3 kN at 50 m to about 27.7–33.3 kN at 500 m, depending on link thickness and trough width. Taking a limiting stress of 400 MPa, the breaking load of the 6 mm chain was 65.5 kN, which gives a factor of safety of 11.2 and 7.4 at 100 m and 150 m for the 250 mm trough and 10.0 and 6.7 for the 300 mm trough. A proposed design that replaces the nut and split dowel pin with a single circlip and shortens the pin reduced the chain mass by 312.6 g per metre (3.96%), reduced the maximum stress in the pin and fastener assembly by 4.2% and its displacement by about 17%, lowered the chain pull by 3.5–3.9%, and gave an improvement of about 4% in the material-stage environmental impact of a 100 m chain. The results provide a calculation-based foundation for chain selection and identify fatigue analysis and field validation as the next steps.

Key Words: Drag conveyor chain; chain pull; finite element analysis; link thickness; factor of safety; circlip; fastener; concurrent engineering; life-cycle assessment.

I. INTRODUCTION

Drag conveyors move bulk materials through an enclosed trough by means of an endless chain that carries projecting flights, and they are widely used for grain, cement, fly ash, and chemical powders. The chain is the element that transmits the drag force to the material, and it works under high tensile load, friction, and sometimes humid or chemical environments. Chain failures such as fracture of the link plates, wear of the pin and bush, and loss of the fasteners cause corrective maintenance and loss of production [1], [2], [3].

The chain considered in this work is a fabricated large-pitch chain of 102 mm pitch used in flat-bottom drag conveyors that convey soybean in oil-processing plants. It consists of outer link plates, inner link plates, bushes, pins, and fasteners, and flights and gussets are welded to the links to drag the material. The outer links are connected through the pins, which are pressed into the outer plates, and the inner links carry the load through the bushes that rotate on the pins. In the existing design, a hexagonal nut screwed on the threaded pin ends and a split dowel pin prevent the outer plate from leaving the pin [4], [5].

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The selection of the link thickness and of the chain for a given conveyor depends on the chain pull, which is the total tensile load on the chain, and on the factor of safety with respect to the breaking load of the chain. The recommended thickness differs between the Japanese and Indian standards, and the stress in the link plates around the holes governs the strength of the chain [6], [7], [8]. Finite element analysis can be used to determine the stress and displacement in the links and in the pin and fastener assembly, and a concurrent engineering approach allows design, manufacturing, and maintenance considerations to be taken into account together during development [9], [10]. This paper reports the design, analysis, and selection of the chain and compares an improved design with the existing design.

II. PROBLEM IDENTIFICATION

Drag conveyor chains work under very severe conditions, and the failures that occur are fracture, cracking, wear, and distortion of the links, pins, and bushes. These failures are attributed to design defects, raw-material defects, operational defects, defects of the fasteners, wrong chain selection, and environmental conditions [1], [11]. In the existing design of the chain, the removal of the nut from the pin due to vibration, forces applied by the outer link, and impact from the sprocket when the chain is misaligned is a major problem, because it allows the outer link to leave the pin, damages the chain, and stops the conveying system.

The problem addressed in this study is to select a link thickness for a 102 mm pitch chain that carries the chain pull of flat-bottom drag conveyors of 250 mm and 300 mm trough width and 100–150 m length with an adequate factor of safety, and to improve the pin and fastener assembly so that it is lighter, simpler to manufacture and assemble, and no weaker than the existing assembly. In addition, the effect of the design change on the chain pull, the stress and displacement of the chain, and the environmental impact needs to be evaluated.

III. RESEARCH OBJECTIVE

The main objectives of the present study are as follows:

- To define the design inputs and the dimensions of the chain components for a 102 mm pitch drag conveyor chain according to IS 6834 and the Japan Chain Industry Association standard.
- To determine by finite element analysis the maximum von Mises stress and displacement in the inner and outer link plates of 5, 6, and 8 mm thickness under tensile loads of 10–50 kN.
- To calculate the chain pull for conveyors of different length and trough width and the breaking and working loads of the chain.
- To select the link thickness for conveyors of 100–150 m length on the basis of the factor of safety.
- To compare the proposed pin and fastener design, which uses a single circlip, with the existing nut and split dowel pin design in terms of mass, stress, displacement, and chain pull.
- To evaluate the manufacturing changes and the environmental impact of the proposed design and to identify the further work required.

IV. MODELING AND METHODOLOGY

4.1 Methodology

A concurrent engineering approach was adopted in which the design of the chain was considered together with its manufacturing, maintenance, and service requirements. The study started with the review of the failures that occur in drag conveyor chains. The chain components were then modelled in SolidWorks. The inner and outer link plates were meshed in HyperMesh using triangular elements of 2 mm size, and a linear static analysis was run with OptiStruct to obtain the displacement and the von Mises element stress, which were viewed in HyperView. The material was taken as steel with a Young's modulus of 2.05×10^5 N/mm², a Poisson's ratio of 0.3, and a density of 7.9×10^{-9} t/mm³, and a tensile force of 10, 20, 30, 40, or 50 kN was applied in the positive X direction. The chain pull was calculated analytically, the breaking load and working load were obtained from the finite element results, and the link thickness

was selected from the factor of safety. The existing and proposed pin and fastener designs were compared by mass, finite element analysis, and chain pull, and the environmental impact of the parts at the material stage was evaluated with the life-cycle assessment software SSLCASoft [12], [13].

4.2 Governing relations

The total chain pull C_p of a drag conveyor is calculated from the Renold designer guide [5] as

$$C_p = 9.81 L [(2.05 W_c \mu_c) + (W_m \mu_m)] + X \quad (1)$$

where L is the centre distance between the drive end and the non-drive end (m), W_c is the total chain mass per metre including attachments (kg/m), μ_c is the coefficient of friction of the chain on steel (0.12 for a 102 mm pitch chain), W_m is the mass of load per metre (kg/m), μ_m is the coefficient of friction of the load on steel (0.40 for grain), and X is the extra chain pull due to side-guide friction, given by

$$X = 2.25 \times 10^4 G L H^2 \quad (2)$$

where G is a factor depending on the material handled (0.05 for grain) and H is the height of the material (m). For a 50 m conveyor of 250 mm trough width with 5 mm thick links, $W_c = 9.56$ kg/m and $W_m = 6.4$ kg/m give $X = 360$ N and $C_p = 2769$ N (2.77 kN), and the chain pull then increases in proportion to the conveyor length.

Because the finite element analysis is linear and static, the maximum stress and displacement in a link are proportional to the applied load. The breaking load of the outer link is taken as the load at which the maximum von Mises stress reaches the limiting stress $\sigma_{lim} = 400$ MPa, which is the lower end of the ultimate tensile strength range of 400–450 MPa taken for steel. The outer link is the only link at the extremes of the chain that receives all the tensile force directly from the pin, so the breaking load of the chain is taken as twice that of one outer link:

$$P_{b,chain} = 2 \times P_{b,outer}, \quad P_{b,outer} = 10 \text{ kN} \times (\sigma_{lim} / \sigma_{10 \text{ kN}}) \quad (3)$$

The factor of safety (FOS) and the working load of the chain are related by

$$FOS = P_{b,chain} / P_w, \quad L_{max} = P_w / c_p \quad (4)$$

where P_w is the working load and c_p is the chain pull per metre of conveyor length obtained from Equation (1), so that L_{max} is the longest conveyor on which the chain can operate at the chosen factor of safety. The relations are preliminary; fatigue, wear, and dynamic loads are not included.

V. DESIGN OF THE DRAG CONVEYOR CHAIN

The chain is assembled from the components shown in Figure 5.1. The dimensions were selected from IS 6834 and the Japan Chain Industry Association standard, which recommend a link thickness of 4.7 mm and 6 mm respectively for the 102 mm pitch, and the link thicknesses of 5, 6, and 8 mm were analysed to cover this range. The materials, principal dimensions, and functions of the components are listed in Table 5.1, and the design inputs used in the calculation are given in Table 5.2.

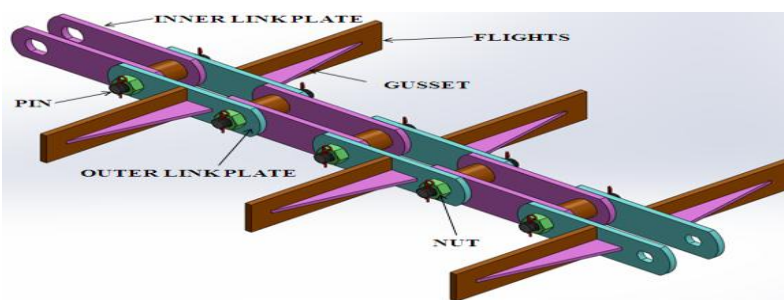


Figure 5.1 Components of the drag conveyor chain

Table 5.1: Components, materials, and principal dimensions of the chain

Component	Material / standard	Principal dimension	Function
Outer and inner link plates	Hot-rolled steel (IS 1079)	5, 6, 8 mm thick; 6 mm in IS 6834	Carry the tensile load
Bush	Case-hardened mild steel	Max. 25 mm dia. (IS 6834)	Bears shear and sprocket impact; bearing for the pin
Pin	Case-hardened mild steel	Max. 15 mm dia. (IS 6834)	Connects inner and outer links
Nut (existing)	ISO 261 coarse thread	M10 × 1.5	Retains the outer link on the pin
Split dowel pin (existing)	Bought-out item	3 mm hole in the pin	Locks the nut
Circlip (proposed)	Bought-out item	Fits pin groove	Retains the outer link on the pin
Flight and gusset	Steel plate	86 mm long for 250 mm trough; 112 mm for 300 mm trough	Drag the material; support the flight

Table 5.2: Design input parameters

Sr.	Parameter	Value
1	Chain pitch	102 mm
2	Trough width	250 mm and 300 mm
3	Conveyor length considered	100–150 m (chain pull evaluated up to 500 m)
4	Material handled	Grain (soybean)
5	Link thicknesses analysed	5, 6, and 8 mm
6	Tensile loads in FEA	10, 20, 30, 40, and 50 kN
7	Young's modulus; Poisson's ratio	2.05×10^5 N/mm ² ; 0.3
8	Friction coefficients	0.12 (chain on steel); 0.40 (grain on steel)
9	Limiting stress	400 MPa
10	Factors of safety considered	2, 4, 6, and 8

In the existing design, the outer link is held on the pin by a nut and a split dowel pin, which requires an external thread and a cross drilling at each pin end. In the proposed design, these two fasteners are replaced by a single circlip, the threading operation is replaced by a simple turning operation, the drilling for the dowel pin is eliminated, and the pin is shortened. Figure 5.2 shows the exploded assemblies of the existing and proposed designs. The existing assembly has six parts, namely the inner link, outer link, bush, pin, nut, and split dowel pin, whereas the proposed assembly has five parts, namely the inner link, outer link, bush, pin, and circlip.

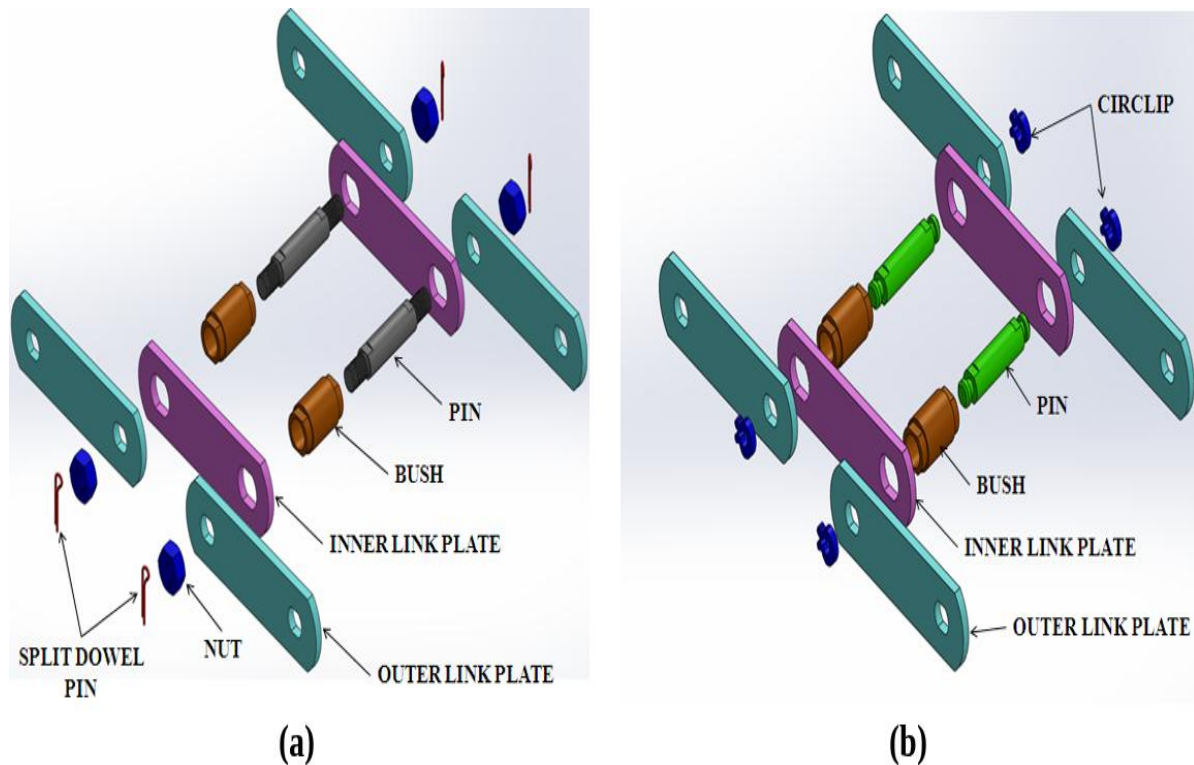


Figure 5.2 Exploded assembly of (a) the existing design and (b) the proposed design

VI. RESULT AND DISCUSSION

6.1 Stress and displacement of the link plates

The maximum von Mises stress and displacement in the outer link are given in Table 6.1 and those in the inner link in Table 6.2. Figure 6.1 shows a typical stress contour of the 5 mm outer link at 10 kN, in which the highest stress occurs around the pin holes, as expected for a tension member with stress concentrations at the holes [14], [8]. Figure 6.2 shows the variation of the maximum stress of the outer link with load.

Table 6.1: Maximum displacement and von Mises stress in the outer link

Tensile load (kN)	5 mm		6 mm		8 mm	
	Disp. (mm)	Stress (MPa)	Disp. (mm)	Stress (MPa)	Disp. (mm)	Stress (MPa)
10	0.041	146.8	0.034	122.2	0.025	91.56
20	0.082	293.7	0.069	244.4	0.051	183.1
30	0.124	440.6	0.103	366.6	0.077	274.5
40	0.165	587.4	0.138	488.8	0.103	366.3
50	0.207	734.2	0.172	611.0	0.129	457.8

Table 6.2: Maximum displacement and von Mises stress in the inner link

Tensile load (kN)	5 mm		6 mm		8 mm	
	Disp. (mm)	Stress (MPa)	Disp. (mm)	Stress (MPa)	Disp. (mm)	Stress (MPa)
10	0.051	191.7	0.043	155.8	0.032	119.0
20	0.103	383.4	0.086	311.5	0.064	238.1
30	0.155	575.1	0.129	467.3	0.097	357.1
40	0.207	766.7	0.172	623.0	0.129	476.1
50	0.258	958.4	0.215	778.8	0.161	595.2

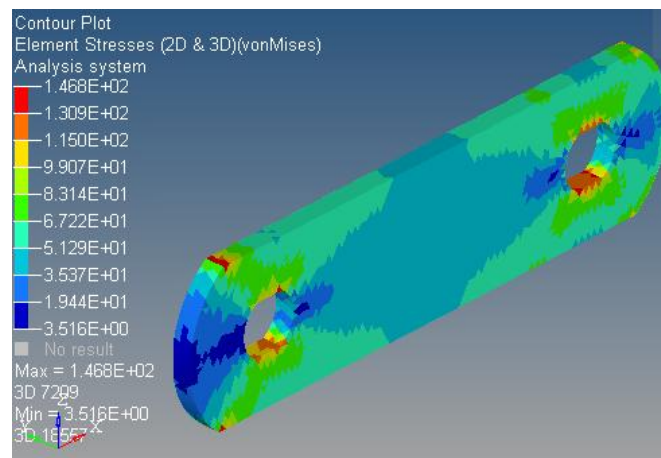


Figure 6.1 Von Mises stress contour of the 5 mm outer link at 10 kN (maximum 146.8 MPa)

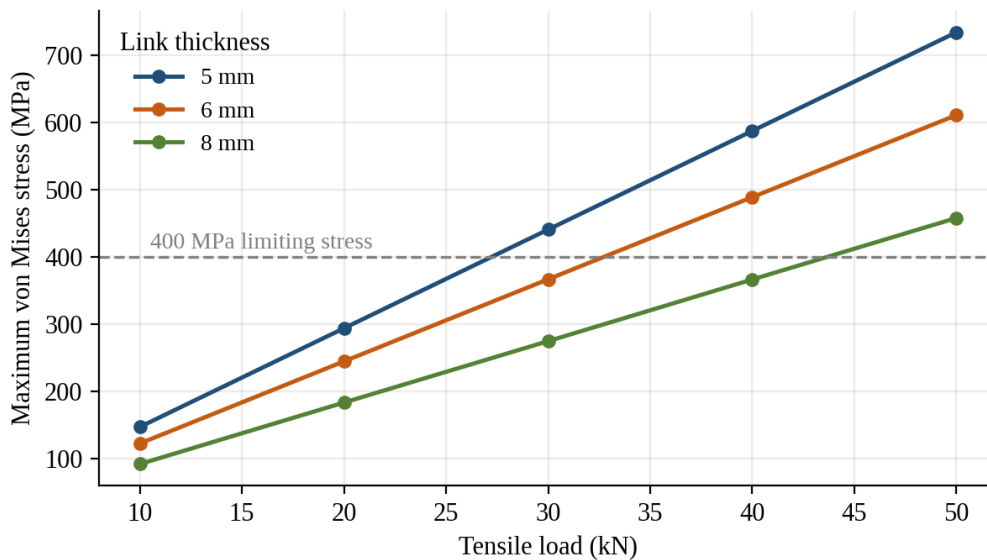


Figure 6.2 Maximum von Mises stress of the outer link as a function of tensile load

Stress and displacement increase linearly with load, as expected for a linear analysis. Increasing the thickness from 5 mm to 6 mm reduces the maximum outer-link stress by about 17% and to 8 mm by about 38%, which is close to the

inverse proportion of the thickness. At the same load and thickness, the inner link carries about 28–31% higher stress than the outer link, for example 191.7 MPa against 146.8 MPa for the 5 mm links at 10 kN. However, the outer link is the only link at the extremes of the chain assembly that receives the tensile force directly from the pin, and the inner link takes the tensile force mainly when the outer links have deformed or failed, so the outer link was used to define the breaking load of the chain. Stresses above the yield strength at the higher loads are indicative only, because the material is assumed linear elastic.

6.2 Chain pull and selection of link thickness

The chain pull calculated with Equations (1) and (2) is given in Table 6.3 for selected conveyor lengths, and Figure 6.3 shows its variation up to 500 m. The chain pull increases in proportion to the conveyor length, and it is higher for the wider trough and for the thicker link because of the larger mass of the chain and the load.

Table 6.3: Chain pull (kN) for different conveyor lengths, link thicknesses, and trough widths

Length (m)	5 mm link		6 mm link		8 mm link	
	250 mm	300 mm	250 mm	300 mm	250 mm	300 mm
50	2.76	3.15	2.93	3.26	3.19	3.33
100	5.53	6.30	5.86	6.53	6.39	6.66
150	8.30	9.45	8.79	9.80	9.58	10.00
200	11.07	12.60	11.73	13.07	12.78	13.33
300	16.61	18.90	17.59	19.61	19.17	20.00
500	27.69	31.50	29.33	32.69	31.95	33.33

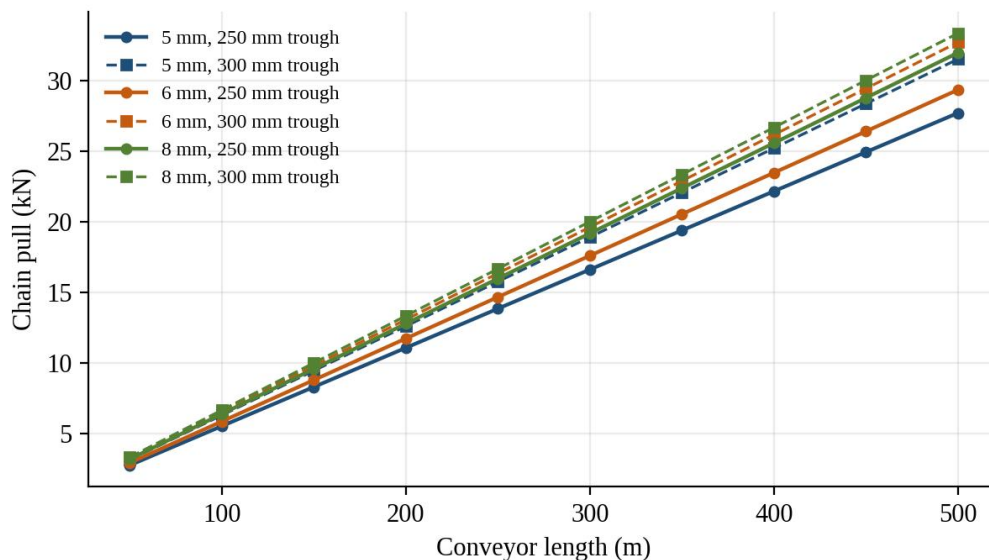


Figure 6.3 Chain pull as a function of conveyor length for 250 mm and 300 mm trough widths

From Equation (3) and the values in Table 6.1, the breaking load of the outer link is 27.2, 32.7, and 43.7 kN for the 5, 6, and 8 mm links, and the breaking load of the chain is 54.5, 65.5, and 87.4 kN. The working load and the longest conveyor on which each chain can operate at a factor of safety of 2, 4, 6, and 8 follow from Equation (4) and are given in Table 6.4, with the length rounded down to the nearest 25 m.

Table 6.4: Working load and maximum conveyor length for different factors of safety

FOS	5 mm link			6 mm link			8 mm link		
	P_w (kN)	L_{250} (m)	L_{300} (m)	P_w (kN)	L_{250} (m)	L_{300} (m)	P_w (kN)	L_{250} (m)	L_{300} (m)
2	27.2	475	425	32.7	550	500	43.7	675	650
4	13.6	225	200	16.4	275	250	21.8	325	325
6	9.1	150	125	10.9	175	150	14.6	225	200
8	6.8	100	100	8.2	125	125	10.9	150	150

L_{250} and L_{300} denote the maximum length for troughs of 250 mm and 300 mm width. For the required conveyors of 100–150 m, the 6 mm link of the IS 6834 thickness gives a factor of safety of 11.2 at 100 m and 7.4 at 150 m for the 250 mm trough, and 10.0 at 100 m and 6.7 at 150 m for the 300 mm trough. A factor of safety of 8 is met by the 6 mm link up to 125 m, and the 8 mm link is required if a factor of 8 is to be kept at 150 m. The 6 mm thickness was adopted for the design because it follows the Indian standard, carries the chain pull of 100–150 m conveyors with a factor of safety above 6 in all the cases considered, and avoids the additional mass and cost of the 8 mm link, which also increases the chain pull.

6.3 Comparison of the existing and proposed designs

The mass of the chain without attachments is 7887.1 g per metre for the existing design and 7574.5 g per metre for the proposed design, a reduction of 312.6 g per metre or 3.96%, as shown in Table 6.5. The reduction comes from the shorter pin and from the replacement of the nut and split dowel pin by the lighter circlip.

Table 6.5: Mass of the chain per metre length (without attachments)

Parameter	Existing design	Proposed design
Parts in assembly	6	5
Fasteners per pin end	Nut and split dowel pin	Circlip
Mass per metre (g)	7887.1	7574.5
Reduction	—	312.6 g (3.96%)

The pin and fastener assembly of both designs was analysed under forces of 5–20 kN applied in the positive X direction, and the results are given in Table 6.6 and Figure 6.4. The proposed assembly gave a lower maximum stress and a lower displacement at every load, with a reduction of 4.2% in stress and 17% in displacement.

Table 6.6: Stress and displacement of the pin and fastener assembly

Force (kN)	Existing design		Proposed design	
	Stress (MPa)	Disp. (mm)	Stress (MPa)	Disp. (mm)
5	149.3	0.0142	143.1	0.0118
10	298.6	0.0285	286.2	0.0236
15	447.9	0.0428	429.3	0.0354
20	597.2	0.0571	572.4	0.0473

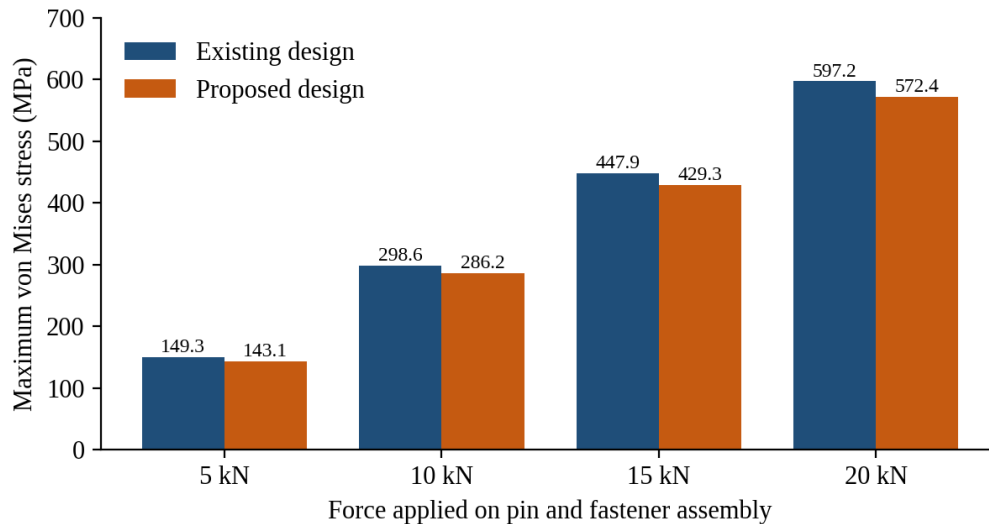


Figure 6.4 Maximum von Mises stress of the pin and fastener assembly in the existing and proposed designs

The lower mass of the proposed chain also reduces the chain pull, because the chain mass per metre appears in Equation (1). Table 6.7 gives the chain pull, maximum stress, and displacement of the 6 mm chain for the two designs, with the proposed design giving a reduction of 3.5–3.9% in chain pull and in stress and a small reduction in displacement.

Table 6.7: Chain pull, stress, and displacement of the existing and proposed 6 mm chain

Trough / length	Existing design			Proposed design		
	Pull (kN)	Stress (MPa)	Disp. (mm)	Pull (kN)	Stress (MPa)	Disp. (mm)
250 mm / 100 m	5.86	71.60	0.020	5.65	69.04	0.019
250 mm / 150 m	8.79	107.41	0.030	8.48	103.62	0.029
300 mm / 100 m	6.53	79.79	0.022	6.28	76.74	0.021
300 mm / 150 m	9.80	119.75	0.033	9.42	115.11	0.032

6.4 Manufacturing and environmental aspects

The proposed design replaces the external threading of the pin ends by a simple turning operation, eliminates the drilling for the split dowel pin, and replaces two fasteners by one standard circlip, which standardizes the assembly, shortens the assembly time, and saves material. The cost of the chain per metre and the manufacturing time are therefore expected to decrease, although these were not quantified in the present analysis. The environmental impact of the parts of the chain at the material stage was evaluated with the SSLCASoft life-cycle assessment software [12], [13]. Human toxicity to air was the dominant impact category, followed by ecotoxicity to water, and the other categories were comparatively small. The proposed design showed an improvement of about 4% in environmental impact for a chain length of 100 m.

6.5 Limitations

The results apply to the conditions studied. The finite element analysis is linear and static, the links are loaded by a uniform tensile force, contact between the pin, bush, and plate is not modelled explicitly, and fatigue, wear, corrosion, and impact from the sprocket are not included, although these are important causes of chain failure [1], [2], [8]. The

breaking load is based on an assumed limiting stress of 400 MPa and not on a tensile test of the chain, the chain pull relation depends on the friction coefficients and material properties assumed, and the environmental assessment covers the material stage only. The results have not been validated by experiments or by service trials, which are necessary before the design is adopted.

VII. CONCLUSION

The design and evaluation of a 102 mm pitch drag conveyor chain for flat-bottom drag conveyors of 250 mm and 300 mm trough width and 100–150 m length were carried out by finite element analysis and calculation of the chain pull. The maximum von Mises stress of the outer link at 10 kN decreased from 146.8 MPa to 122.2 MPa and 91.56 MPa as the link thickness increased from 5 mm to 6 mm and 8 mm, and the inner link carried about 28–31% higher stress. With a limiting stress of 400 MPa, the 6 mm chain has a breaking load of 65.5 kN and a factor of safety of 11.2 and 7.4 at 100 m and 150 m for the 250 mm trough, and 10.0 and 6.7 for the 300 mm trough, so it was selected for the design. Replacing the nut and split dowel pin by a single circlip and shortening the pin reduced the chain mass by 312.6 g per metre (3.96%), the stress in the pin and fastener assembly by 4.2%, its displacement by about 17%, and the chain pull by 3.5–3.9%, simplified the manufacturing and assembly, and improved the material-stage environmental impact by about 4%. Further work should include fatigue and contact analysis of the links and pin assembly, tests of the chain, field trials, quantified cost analysis, chains of other pitches, and a complete life-cycle assessment.

REFERENCES

- [1] Renold Power Transmission, *Renold Troubleshooter Chain Guide*, 2012.
- [2] N. I. B. Haris, "Failure analysis of conveyor chain links: A case study at Top Glove Sdn. Bhd.," M.Eng. project report, Universiti Tun Hussein Onn Malaysia, July 2013, pp. 1–55.
- [3] S. M. Bošnjak *et al.*, "Failure investigation of the bucket wheel excavator crawler chain link," *Engineering Failure Analysis*, vol. 35, pp. 462–469, 2013.
- [4] S. Shimura, *The Complete Guide to the Chain*, 1st English ed. Tsubaki, 1997, pp. 1–237.
- [5] Renold, *Conveyor Chain Designer Guide*, Sec. 4, pp. 66–104.
- [6] Japan Chain Industry Association, Japan Chain Industry Association Standard JCAS 2–1982, 1982.
- [7] Bureau of Indian Standards, *Indian Standard Specification for Conveyor Chains, Chain-Wheels and Attachments, Part I*, IS 6834 (Part I)–1973, reaffirmed 2010.
- [8] M. D. Jagtap, "Study of roller conveyor chain strip under tensile loading," *International Journal of Modern Engineering Research (IJMER)*, vol. 4, pp. 61–66, May 2014.
- [9] S. M. Bogus *et al.*, "Concurrent engineering approach to reducing design delivery time," *Journal of Construction Engineering and Management*, pp. 1179–1185, Nov. 2005.
- [10] T. D. Bhoite, "FEA based study of effect of radial variation of outer link in a typical roller chain link assembly," *International Journal of Mechanical and Industrial Engineering*, vol. 4, pp. 65–70, 2012.
- [11] TCR Engineering, "Failure and root cause analysis," *TCR Engineering Investigating Material and Component Failure*, July 2004, pp. 1–30.
- [12] S. Sharma, "Investigation of engineering products for environmental impact evaluation through life cycle assessment: Two-wheeler and some other consumer products," Ph.D. dissertation, Oct. 2012, ch. 3.
- [13] S. Sharma, "Environmental impact evaluation: An investigation of cylinder assembly and cylinder head assembly of engine group," *International Journal of Engineering Science and Technology*, vol. 4, pp. 3200–3203, July 2012.
- [14] M. D. Jagtap, "Use of strain gages to analyze the behavior of roller conveyor chain strip," in *Proc. Int. Conf. Computer Science and Mechanical Engineering*, Aug. 2014.