A CASE STUDY OF THE ROLLER CHAIN STRENGTH AND RELIABILITY OF SUGARCANE MILL CONVEYING SYSTEMS

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ABSTRACT:

In the sugar industry, different types of chains play a vital role in transporting materials like sugarcane, bagasse, and sugar. These chains face various loads, and when they fail, it disrupts sugar mill operations, leading to production stoppages and costly repairs. Strengthening the chains to handle these loads is crucial for keeping operations smooth and cost-effective.

Hence, the selection of the chain is crucial for engineers, and should consider special features such as robust construction, high tensile strength, durability, resistance to acidic juice, etc.Thus, it emphasizes state-of-the-art technology, material construction, and selecting the appropriate chain essential to achieving maximum productivity, improving drive service life, and reducing long-term costs in sugar mills.

The authors focus on studying the selection of the type of chain, the material of construction, design, and manufacturing processesof chains. The study starts with theoretical analyses of different roller chain parts, like pin, bush, andlink, to see how they handle breaking loads. To double-check these findings, finite element analysis (FEA) is done. The FEA results confirm the stresses these components experience.

The study finds/shows acceptable roller bush stresses, but roller pins and links have excessively high-stress levels, posing a risk of failure under breaking loads. To address this issue, the study suggests using superior materials for roller pins and links.

The study utilizes both theory and Finite Element Analysis (FEA) to assess the strength of the roller chain components in the carrier system of the sugar industry. The objective is to identify weak points and propose methods to reinforce them, ultimately leading to cost savings on repairs and prolonged lifespan of the handling system chains.

Keywords: Structural Analysis, Roller Conveyor, Theoretical Calculations, FEA Analysis, Von Mises Stresses

INTRODUCTION:

The chain is the most important, reliable, and critical machine component in sugar mill operation, which transmits power by means of tensile forces.It is used in conveyors for cane, bagasse, etc. Most of the time chains are subjected to extreme service conditions like high tensile loads which cause elastic and plastic stresses resulting in elongation of the chain. Sugar mills utilize various types of chains like roller-bush-pin type for cane carriers, feeder tables, bagasse conveying, and block forged type for prepared cane and juice-contained bagasse, duplex/triplex chains for UFR, TRPF, and GRPF, as the main conveying system and torque transmission. As these chains operate under various loads, failure of chain assembly is a perennial problem, disruptingthe operation of a sugar mill. Hence, the selection of a chain is one of the prime factors for engineers.

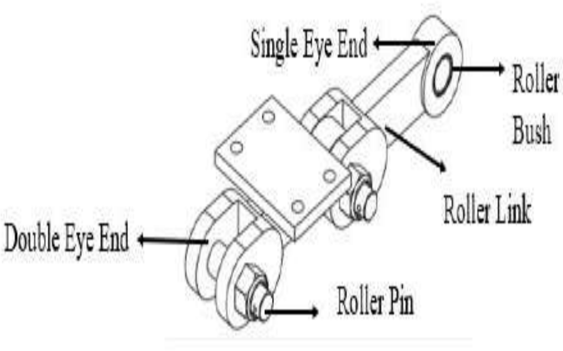
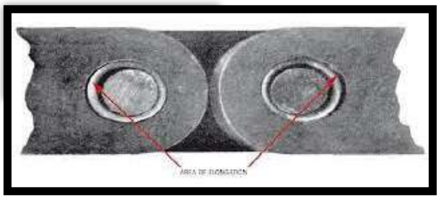


Fig 1: Inter-rake Carrier roller chain Assembly.

Chainsplay vital roles in the sugar industry for efficient material handling and transportation. Conveyor chains are crucial for transporting cane and bagasse. Cane carrier chains facilitate the movement of harvested cane, while bagasse carrier chains transport bagasse. These chains are designed to withstand harsh conditions, resist wear, and ensure smooth flow.Conveyor chains are meticulously manufactured with appropriate materials and treatments to improve their strength and corrosion resistance.Chains contribute to the overall efficiency of sugar production by facilitating seamless material flow.

Table 1: Some problems/failures observed for various conveying chains

	Particulars	**Problems/Failure Observed**
	Roller Jamming	• Due to the ingress of bagasse and stick in the gap of the inner link, the pin and roller • Rollers do not rotate freely and continuously rub on the runner.
	Link Plate Wear	• After roller jamming, the roller gets worn out and link plates start contacting with the runner and get worn out.
	Excessive noise	• Caused by broken links or chain rollers, extrememisalignment, elongation, and chain jumping over sprocket teeth.

	Chain wear and elongation	• Normal wear which will increase chain elongation and cause severe wear on the tips of the sprocket teeth.
	Sprocket wear	• Due to the worn-out sprocket, the chain tends to cling to the sprocket teeth and vibrate.
	Chain parts broken	• Caused due to overload,misalignment, excessive elongation, and weak tensile strength.

<u>Section criteria for parts of a roller chain with their function:</u>

Chain Dynamics and Selection of Chain:

- Selection of chain pitch and chain pull calculation.
- Factor of safety:
  Inthe most common application, a factor of safety of a minimum of 8 (eight) shall be maintained.
  Working Load = Breaking Load / 8
- Expected Chain life in working hours.

Material of Construction:

- Inter-rake carrier chains transfer bagasse in sugar production, resisting juice corrosion.
- For this application, use block-type links without rollers and have a breaking strength of 20,000 - 130,000 kg.
- The chain links are forged with low- or medium-carbon steel, while pins and bushes undergo heat treatment to prevent elongation.
- Stainless steel components enhance durability and corrosion resistance.
- Designed for highly corrosive environments, these chains reliably operate under continuous exposure to raw sugar juice.

a) Chain links:

- Carbon steel has intuitive strength.
- High-precision machining to achieve ideal tolerance minimizing friction.

- Ensuring requisite harness by appropriate heat treatment process/ method to endure dynamic friction, wear and tear.

b) Pins:

- Low carbon alloy steel / Stainless steel (Austenitic – SS 400 series (SS 410, SS420, SS432) & Martensitic - SS 300 series 9SS304, SS310, SS316)).
- Ideal tolerance by precision machining. Suitable Heat treatment like Case carburizing, hardening, and tempering process.

c) Bushes:

- Alloy steel.
- Precise machining to achieve smooth outer and perfect tolerance.
- Suitableheat treatment extra strength to resist vigorous friction and wear & tear.

Manufacturing Process:

The following are the engineering procedures adopted by chain manufacturers.

- Raw material: Purchased from a reputed manufacturer and established with controlled chemistry of the material.
- Steel from process route EAF/LRF/VD/EMS/AMLC is specified to get the desired results from heat treatment of the material.
- Heat treatment - Quench gas carburizing.
- Hardening and tempering process and Oil and polymer quenching.

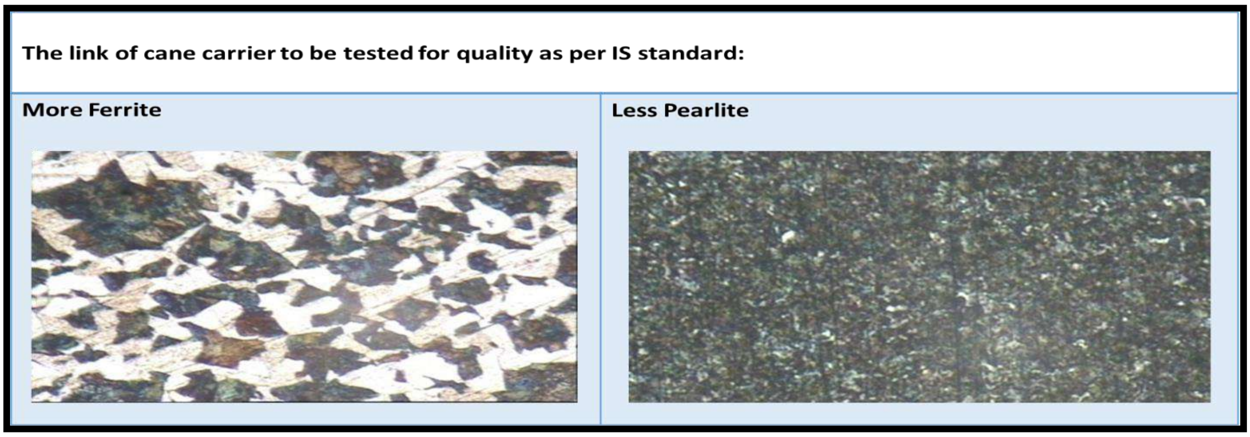


Fig. 2: Microstructure of more FerriteandLess pearlite materials

- The material with more ferrite and less pearlite indicates that links were manufactured without heat treatment. The same links were also checked with heat treatment, and the microstructure was found to be the same as above.
- Less free ferrite indicates more tensile strength, reducing the chances of chain link elongation and breakage.
- Meticulous attention should be given to the quality of the chain material, manufacturing process, and quality checks.

- The quality of the chain depends on both mechanical properties, such as tensile strength, breaking load, % elongation, wear and tear, and chemical properties, such as corrosion resistance. This is especially important for chains that come into contact with acidic cane juice.
- The chains shall be specially engineered considering the rugged applications in sugar mills.
- Thus it is important to study the strength and operation of the chain, which governs the sugar mill operation.

Theoretical Analysis of Roller Chain Parts:

For further diagnosis of the current issues related to material composition, design, selection, manufacturing processes, and testing procedures of chains, a theoretical analysis of roller chain parts was conducted by calculating the Von-Mises stresses.

The Von-Mises theory, also known as the maximum distortion energy theory, is widely used to analyze yielding and failure in ductile materials. This theory takes a conservative approach by considering distortion energy as the criterion for yielding. It is more conservative in its estimations compared to other theories. On the other hand, the Tresca theory, also known as the maximum shear stress theory, focuses solely on shear stress and does not take into account the impact of hydrostatic stress. It is better suited for materials sensitive to shear stress rather than hydrostatic stress. The Mohr-Coulomb theory combines shear strength and cohesion in its analysis of shear and compressive failure. It is typically applied to materials exhibiting significant plastic deformation and shear failure.

The Von-Mises theory stands out for its ability to offer a more accurate depiction of yielding in ductile materials, as it takes into account both shear and hydrostatic stress components. This theory finds widespread usage in engineering applications involving ductile materials, such as metal structures. However, it may not entirely capture the failure behaviour in scenarios where pressure-sensitive yielding or significant plastic anisotropy is involved. In such cases, alternative yield criteria or advanced constitutive models may be required to provide more accurate predictions.

$\sigma_v = \sqrt{(\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 + 3\tau_1^2 + 3\tau_2^2 + 3\tau_1\tau_2)}$

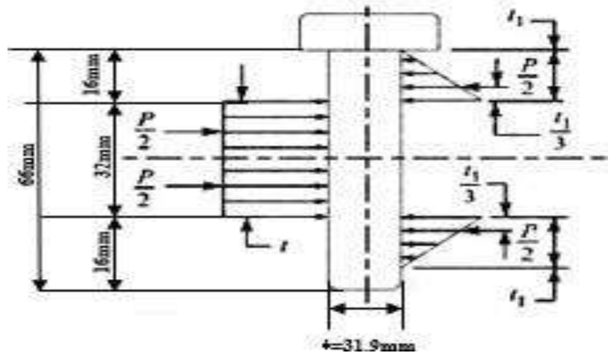
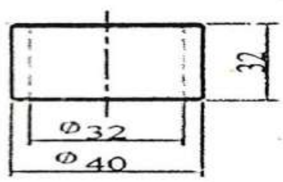
where: σ_v is the Von-mises stress, σ_1 and σ_2 are the principal stresses, τ_1 and τ_2 are the shear stresses.

Table 2: Physical properties of the material

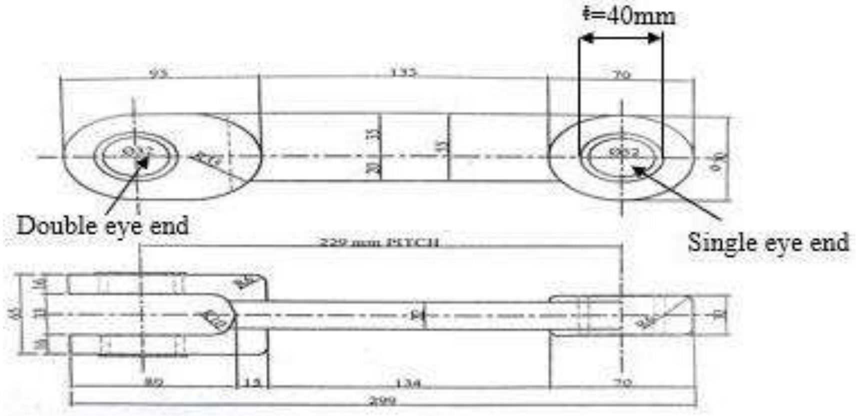
Particular	Unit	Pin	Bush	Link
Various parts of roller chain				
Material		SS 410	SS 304	CarbonForgedSteel
Modulus of Elasticity	GPa	200	193	200

Density	g/cc	7.8	8.00	7.8
YieldStrength	MPa	290	215	325
Ultimatestrength	MPa	510	505	625

Table 3: Design of RollerChain Pin, bush, and link

<table>
<tr><td colspan="3">
</td></tr>
<tr><td>Breaking Load: 60000Kg
P = 588.6kN
For Two Strand:
P = 588.600/2 = 294.300 kN
P= 294300 N
Possible failure in Roller Pin:</td><td>Bending Stress:
Bending Moment = M = P/2(t1/3+t/2)-(P/2 * t/4)
M = (294.3*103)/2(32/3+16/2)-((294.3*103)/2 * 16/4)
M = 2157219 N.mm
Section Modulus =Z = (π /32 *(d13))
Z = (π /32 *((32)3))
Z = 3216.99 N/ mm2
Maximum Bending stress =
σb = M/Z
σb = 2157219/3216.99
σb= 670.57MPa
Shear stress: Crushing Stress:
ح= P/(2*(π /4)*(d1^2) =182.96 MPa
σCrush = P/(bd) = 139.35MPa</td><td>Von-Misses stresses:
σv = √σ ^{2}-(σx*σy) + σy2+3حxy2
σx = σb + σc = 670.57+139.35= 809.92MPa
σy = σz =0
حxy2 = 182.96 MPa
σv = √(809.92)2 - (809.92*0) + (0)2+3(182.96)2
σv = 869.70 MPa
Factor of Safety:
F.O.S = Sut/ σv
= 510/869.70
F.O.S = 0.5864 < 1</td></tr>
<tr><td colspan="3">
</td></tr>
<tr><td colspan="2">Crushing Stress:
σCrush = P/(bd)
σCrush = (294.3*103) / (32*32) σCrush = 287.4 MPa

Checking for von-Misses stresses in Bush.</td><td>Von-Misses stresses:
σv = √σx2-(σx*σy) + σy2+3حxy2
σx = σc = 287.4 MPa
σy = σz =0
حxy2 = 0
σv = √(287.4)2-(287.4 *0)+ (0)2+3(0)2
σv = 287.4 MPa
Factor of Safety:</td></tr>
</table>

	F.O.S = Sut/ σ_v = 505/287.4 F.O.S = 1.75 > 1
 	
Possible failure in Roller Link: Tensile Failure: $\sigma_t = P/(d_{3o}-d_{3i}) * t$ = 306.56MPa Shear Failure: $\tau = P/((d_{3o}-d_{3i}) * t$ = 306.56MPa Crushing Failure: $\sigma_{Crush} = P/(d_{3i} * t)$ = 229.92MPa	**Von-Misses stresses:** $\sigma_v = \sqrt{\sigma_x^2-(\sigma_x*\sigma_y) + \sigma_y^2+3\tau_{xy}^2}$ $\sigma_x = \sigma_b + \sigma_c$ = = 536.48MPa $\sigma_y = \sigma_z = 0$ τ_{xy}^2 = 306.56 MPa $\sigma_v = \sqrt{(536.48)^2 -(536.48*0)+ (0)^2+3(306.56)^2}$ σ_v = 754.81 MPa Factor of Safety: F.O.S = Sut/ σ_v = 625/754.81 F.O.S = 0.82 < 1

a) Numerical Model and Mathematical Formulation

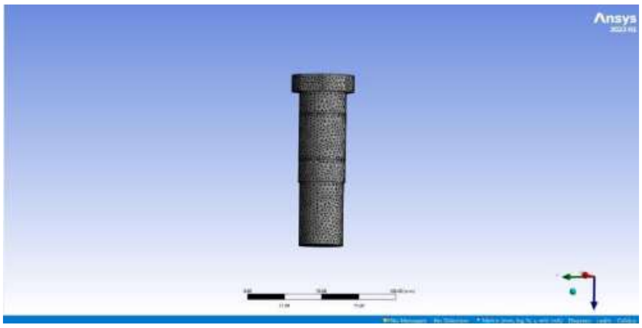
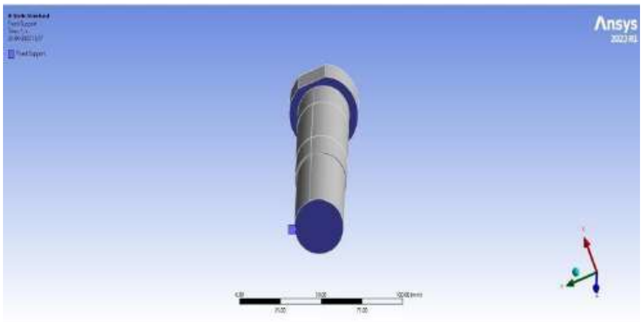
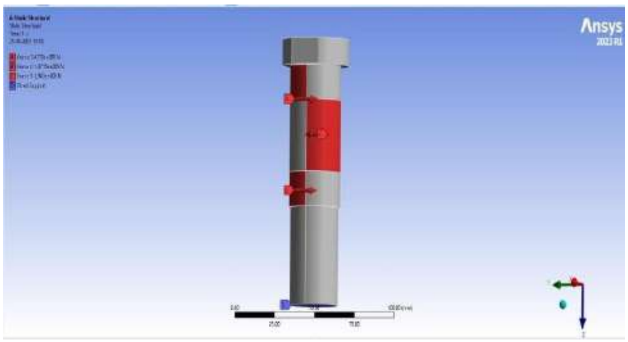
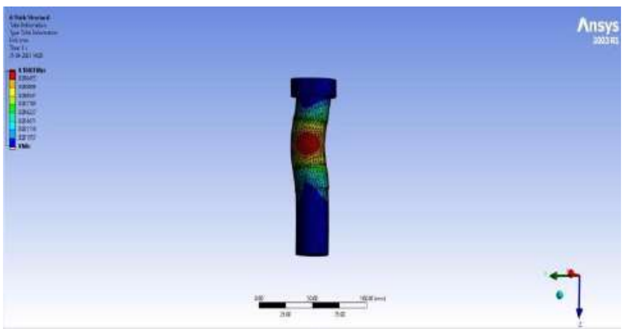
To assess the breaking strength of roller chains used in the sugar industry and suggest suitable materials, heat treatment (if necessary), and specific manufacturing processes (if needed) for various chain parts, such as roller pins, roller bushes, and links. In addition, a life assessment of roller pins and bushes will be conducted to determine damage tolerance through wear analysis.

The study will commence with a theoretical analysis of various chain parts, such as roller pins, roller bushes, and roller links, to evaluate their ability to withstand breaking loads. To validate these findings, finite element analysis (FEA) will be used. The FEA results will confirm the stresses experienced by these components.

In addition, the project underscores the importance of conducting a thorough analysis of the mechanical properties and material availability to select appropriate materials for the roller chain components. This involves evaluating the wear rate and coefficient of friction between the roller pin and bush materials across different operating conditions.

b) Finite Element Analysis of Roller Chain:

Table 4: Meshed Model of Roller Pin

	Roller Pin Meshing: In order to carry out the stress analysis, mesh was developed for the Roller Pin. The mesh consists of 58333 nodes and 83926 elements and element size of 2.5 mm. The meshing of the domain is as shown.
	Boundary Condition 1: The Roller Pin fixed one end at point c.
	Boundary Condition 2: The axial tensile force of 294300N is applied on the surface.
	Results: Roller Pin Deformation: Tensile load applied. Max deformation: 0.1040 mm, Graphical representation highlights zones, Red: Max deformation, Blue: Min deformation.

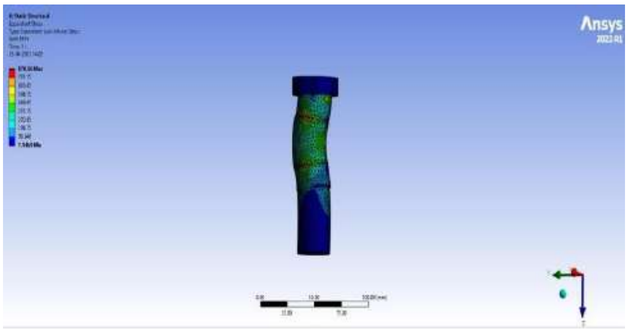
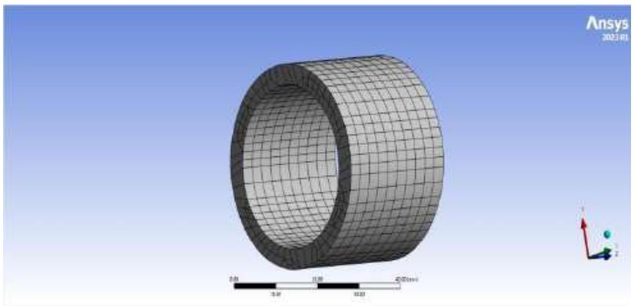
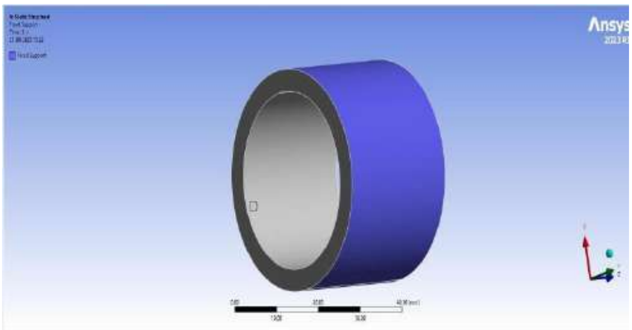
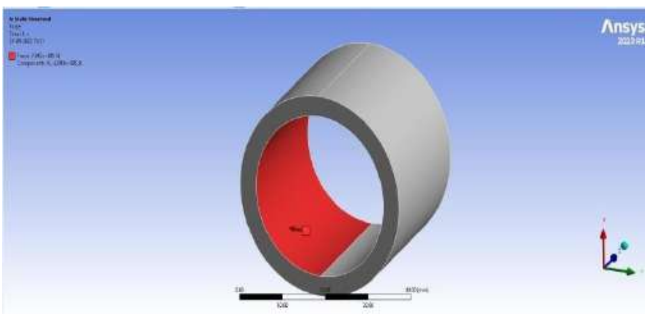
	Roller Pin Total Deformation Equivalent Stresses: Fig shows equivalent stresses produced in the Roller Pin. The maximum tends to be 878.66 MPa while the minimum varies to 1.1469MPa.

Table 5: Meshed Model of Roller Bush

	Meshed Model of Roller Bush: In order to carry out the stress analysis, mesh was developed for the Roller bush. The mesh consists of 87644 nodes and 20536 elements and element size of 2.5 mm. The meshing of the domain is as shown
	Boundary Condition 1: The Roller Bush fixed the outer surface at point c.
	Boundary Condition 2: The axial tensile force of 294300N is applied on the inner surface.

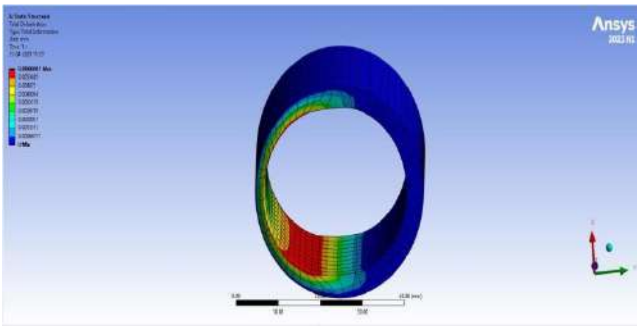
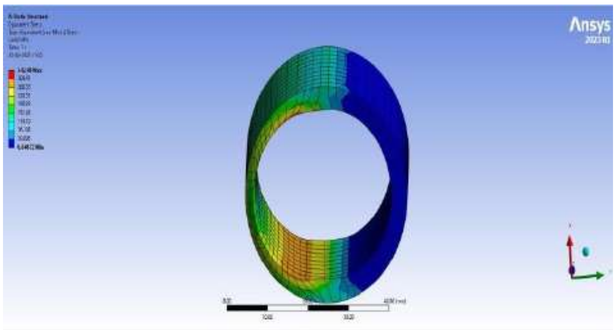
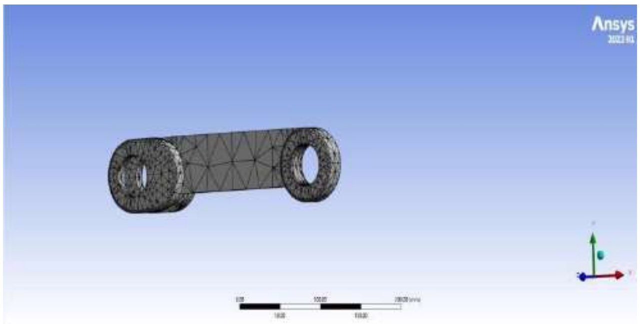
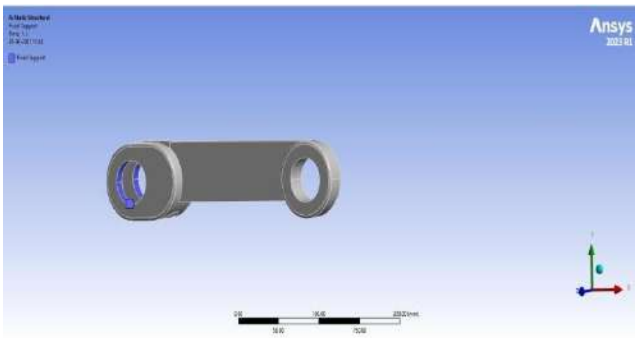
	Results indicate that under tensile load, the roller pin experiences a maximum deformation of 0.0060081mm. A graphical representation highlights different zones, with red indicating maximum deformation and blue representing minimum.
	Equivalent Stresses: shows equivalent stresses produced in the Roller Pin. The maximum tends to 342.48 MPa while the minimum varies to 0.04872MPa.

Table 6: Meshed Model of Roller Link

	Meshed Model of Roller Link: In order to carry out the stress analysis, the mesh was developed for the Roller bush. The mesh consists of 50975 nodes and 32493 elements and element size of 2.5 mm. The meshing of the domain is as shown.
	Boundary Condition 1: The Roller Link is fixed at one end at point c.

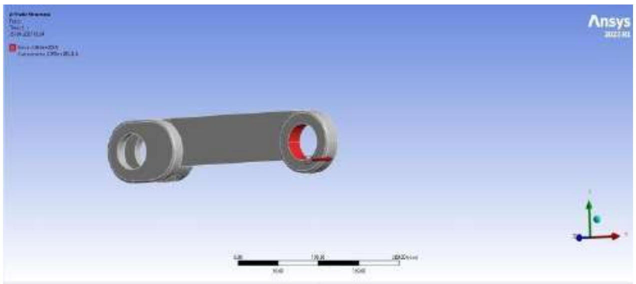
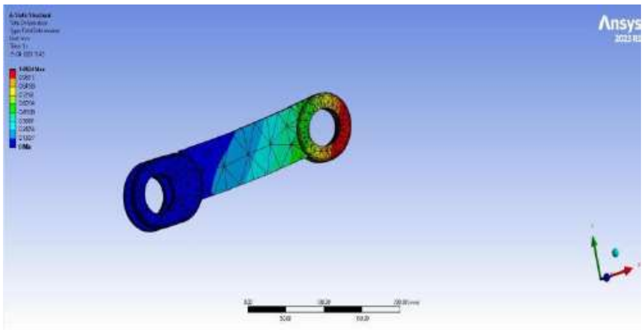
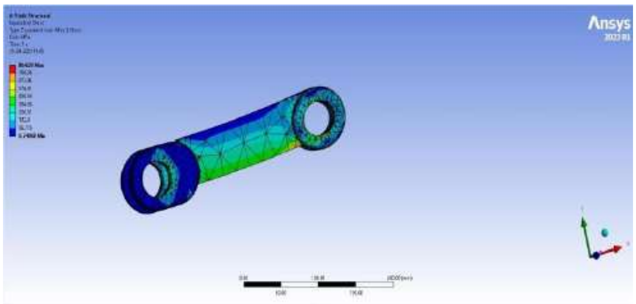
	Boundary Condition 2: The axial tensile force of 294300N is applied on the inner surface.
	Deformation of Roller Link: After applying tensile load in the roller link it is seen that the maximum deformation occurs is 1.0824 mm as shown. Fig shows the graphical deformation that occurred and various zones are highlighted in different colours. Red indicates maximum deformation while the blue region shows minimum.
	Equivalent Stresses: Shows equivalent stresses produced in the Roller Pin. The maximum tends to be 864.99 MPa while the minimum varies to 0.74868 MPa.

Table7:ComparisonResults

Part No.	Part Name	Existing Material & its Strength (Sut MPa)	Theoretical ((σv)MPa)	FEA (σv MPa)	F.O.S= Sut/σv	Deviation (%)	Remark
1	RollerPin	510	869.70	878.66	0.58	1.01	Not safe
2	RollerBush	505	287.4	342.48	1.75	16.05	Safe
3	Roller Link	625	754.81	864.99	0.82	12.73	Not safe

The pin and link material are not safe against the maximum load because the factor of safety is less than 1. To achieve an optimum factor of safety of 1, we need to determine the required yield strength of the potential material for the pin and link. However, the bush material is safe against the maximum load as the factor of safety is greater than 1.

Failure analysis of roller chain parts using theoretical and FEA techniques. Suggest appropriate material of construction (if required), propose suitable heat treatment (if required), and conduct wear analysis of the roller pin using experimental techniques.

The table above compares theoretical and finite element analysis (FEA) results for three different parts: roller pin, roller bush, and roller link. It compares theoretical stress (σv) values with FEA stress (σv) values and calculates the percentage deviation between the two. This is important for assessing the accuracy of theoretical models and validating FEA simulations used to analyze the structural integrity of these components.

In the case of the roller pin, the theoretical stress value is 869.70 MPa, while the FEA stress value is slightly higher at 878.66 MPa, resulting in a deviation of 1.01%. Such a small deviation indicates that the theoretical model used to calculate the stress on the roller pin is reasonably accurate. The FEA results validate the theoretical model, confirming that the pin can not handle the expected load without significant discrepancies.

The roller bush exhibits a deviation of 16.05%. The theoretical stress is 287.4 MPa, while the FEA stress is significantly higher at 342.48 MPa. Similarly, the roller link shows a 12.73% deviation between theoretical and FEA stress values. The theoretical stress is 754.81 MPa, and the FEA stress is 864.99 MPa.

Since the roller carrier chain is a standard part of the carrier assembly, the dimensions of the components are to be maintained constant. For analysis, the rated load is taken into consideration for simulation. Thus, to bring the design within the safety threshold, changes have to be made to the material used. Hence, new materials can be suggested or changes in current materials are to be made.

a) Roller Pin:

- Stainlesssteel 410 is a commonly used material for roller pins, but in this case, the strength properties are not sufficient to carry out the performance.

- Stainless steel 410 is a martensitic stainless steel that is often used in applications requiring moderate corrosion resistance and high strength. The ultimate tensile strength of stainless steel 410 is around 510 MPa, which is lower than the minimum requirement of 870 MPa specified in the question.

- For roller pin applications, suitable material would generally require higher strength and toughness. However, it is important to consider other factors, such as corrosion resistance, cost, and availability, when selecting a material. Stainless steel 410 may still be a viable option if corrosion resistance is a critical factor and the load requirements are lower than 60000 N.

- For roller pin Von-Mises stresses, they are 878.66 MPa as per FEA, so the suitable material should have an ultimate strength greater than approximately 880 MPa. The design will sustain the applied load in such a case where FOS is considered 1. For FOS 1.2 and 1.5, the yield strength should be greater than 1156 MPa and 1320 MPa, respectively.

b) Roller link:

- Forged steel is a commonly used material for roller link applications, but in this case, the strength properties are not sufficient to carry out the performance.

- Carbon-forged steel is a type of carbon steel that has been forged to improve its strength and durability. The ultimate tensile strength of carbon-forged steel is around 625 MPa, which is lower than the minimum requirement of 755 MPa.

- Carbon-forged steel can be heat-treated to achieve even higher strength and toughness.

- For roller link applications, the preferred material should have higher strength, toughness, and fatigue resistance. However, it is important to consider other factors, such as cost and availability, when selecting a material. Carbon-forged steel may still be a viable option if the load requirements are lower than 60000 N and the cost is a critical factor.

- For roller links, Von-Miss stresses are 754.81 MPa as per calculation and 864.99 MPa as per FEA, so the suitable material should have a yield strength greater than approximately 865 MPa. The design will sustain the applied load in such a case where FOS is considered 1. For FOS 1.2 and 1.5, the yield strength should be greater than 1038 MPa and 1297.5 MPa, respectively.

CONCLUSION:

- The initial materials of pins (stainless steel) and block-type links (forged steel) failed to meet the design requirement as per theoretical and FEA analysis. Therefore, an analysis of mechanical properties and material availability was conducted to determine alternative materials.
- Based on the findings, it is recommended to replace stainless steel pins with AISI 4140 OR SS 410 tempered at 204°C (1000 MPa) and to replace forged steel links with AISI 4340 OR C 55 Cr 75 hardened & tempered (900 - 1050 MPa) in future designs.
- This project has shown that a thorough analysis of mechanical properties and material availability is necessary to identify alternative materials that can meet the design criteria.
- The wear rate in our experiment for stainless steel 410 is also between $1*10^{-6}$ and $5*10^{-5}$ mm3/Nm, which is the general range of the wear rate of stainless steel. So it is important to focus on factors that influence the wear rate.
- The coefficient of friction of SS410 can vary depending on the conditions and the counter material it is in contact with. However, as a general range, the coefficient of friction of SS410 can be between 0.2 and 0.6. In our experiment, the coefficient of friction obtained ranged from 0.20 to 0.28.

ACKNOWLEDGEMENT:

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