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OPTIMIZATION OF WEIGHT OF ROLLER CHAIN INNER LINK PLATE FOR TYPICAL INDUSTRIAL CHAIN APPLICATION

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ABSTRACT

Optimization is the process of obtaining the best result under given circumstances in design of system. In optimization process, we can find the conditions that give the maximum and minimum value of function. In this study, a weight optimization process is used for the design of roller chain link for minimization of failure modes. This process has various design variables, such as wall thickness of link, breaking area of link, inner width of chain and shape of the link. While deciding the optimized weight of the roller chain link, raw material plays important role, so it is necessary to decide raw material. Normally medium alloy steel i.e. as per Indian Standard C45, 55C8 or as per British Standard EN19 has been used in normalized condition and after manufacturing of link it has been heat treated up to 35 to 40 HRC in order to get tensile strength up to 70 to 80 kg/mm².

KEYWORDS: Chain Links, Ansys, Finite Element analysis, Weight optimization of chain links etc.

INTRODUCTION

India is very rich in its varied heritage in different fields. India's economy is depending on two major sectors i.e. industrial and agricultural. The major process in any industry depends of the transmission of material through conveyor. For transferring the power over long distance conveyor can be very useful to reduce manpower cost and lower material handling. But to accomplish this precise task we need to transfer mechanical output of the motor to conveyor roller by using chain. It is surveyed, and based on cast experience of author majority of failures in industries are because of chain links. Mostly causes of failure are improper design, improper material selection for chain making, and lack of awareness about chain tension or fitting of the chain, this causes the failure of the chain.

DESIGN PROCESS BASED ON BREAKING LOAD

The breaking load is calculated by using below formulae.

Torque on driven / driver pulley P_d:

$T_1 -$ Torque on Driver sprocket (Nm)
$T_2 -$ Torque on Driven sprocket (Nm)
$P_1 -$ Driver power Transferred (kW)
$\eta -$ Efficiency (normally about 98%)
$n_1 -$ Driver sprocket rotational speed (rpm = m^{-1})
$n_2 -$ Driven sprocket rotational speed (rpm = m^{-1})
If the input power = P_1 (kW) then the torque (Nm)
For Driver Sprocket,

$$T_1 = \frac{P_1 \times 9.549}{n_1}$$

$$T_1 = 37.32\ Nm$$

For Driven sprocket,

$$T_2 = 29.74\ Nm$$

So as per the above analytical calculations we got torque of driver sprocket is 37.32 Nm and driven sprocket if 29.74 Nm.

i. Chain Velocity v:

For driver sprocket,

$$v_1 = \frac{D_1 \times \pi \times n_1}{60}$$

$$v_1 = 0.142\ m/s$$

For driven sprocket,

$$v_2 = \frac{D_1 \times \pi \times n_2}{60}$$

$$v_2 = 0.138\ m/s$$

So as per the above analytical calculations we got chain velocities driver sprocket side velocity is 0.142 m/s and driven sprocket side velocity if 0.138 m/s. Now by using that working stress values we calculate the working load the chain link plate can carry by using the following formulae.

SAMPLE CALCULATION DESIGN POWER FOR ROLLER CHAIN:

$$P_d = P \times K1 \times K2 \times K3 \times K4 \times K5 \times K6 \times K7$$

Given,
$P_d -$Preliminary chains pull (kW)
$K1 -$Coefficient for teeth different to 19
$K2 -$Coefficient for Transmission Ratio
$K3 -$Application (Service) Factor
$K4 -$ Centre Distance Coefficient
$K5 -$Lubrication Coefficient
$K6 -$ Temperature Coefficient
$K7 -$ Service Life Coefficient (For coefficients refer Annexure III)

$$P_d = 19.543\ kW$$

So as per the above analytical calculations we got chain design power of chain which is 19.543 kW.

ii. Chain Link Plate Power Capacity:

$$P_L = (K_a \times Z_1^{1.06} \times n_1^{0.9} \times P_i^x) \times 0.7457$$

Given,
P_L – Link power capacity (kW)
Z_1 –Driver Sprocket- Number of teeth
n_1 – Driver Sprocket rotational speed (rpm)
P_i –Pitch of the chain (inches)
K_a –Link power factor

$$P_L = 5.48\ kW$$

So as per the above analytical calculations we got chain link plate power capacity which is 5.48 kW.

iii. Tensile load on chain:

$$F_t = \frac{P_1 \times 1000}{v}$$

$$F_t = 13.76\ kN$$

iv. Breaking load of Roller Chain:

$$Breaking\ load = \frac{F_t \times 8}{2}$$

$$Breaking\ load = 55.05\ Tons \approx 55\ Tons$$

By using the survey data author have been found the breaking load of 55 Tons. This breaking load has been calculated at the joint of power transmission roller chain in this joint there is need to find out minimum cross-sectional area for both outer and inner link plate and diameter of pin subjected to shear and bending stresses.

Calculations for Chain link plate:

As per the catalogue we had taken chain link plate of EN-19 material, dimensions 68.30 mm x 76.20 mm (Pitch) x 9.5 mm for outer link plate and 68.30 mm x 76.20 mm (Pitch) x 11.30 mm for inner link plate. Now by using the analytical formulae we find out the value of maximum stress i.e. ultimate tensile strength. Values from design data book [40].
Tensile strength = 1097 N/mm^2 to 1231 N/mm^2
Modulus of elasticity = $2.05X10^5\ N/mm^2$
Poisson's ratio = 0.3
Maximum working stress-

$$Maximum\ Working\ stress = \frac{Maximum\ stress}{Factor\ of\ safety}$$

$$Maximum\ Working\ stress = 821.45\ N/mm2$$

Minimum working stress-

$$Minimum\ Working\ stress = \frac{Minimum\ stress}{Factor\ of\ safety}$$

$$Minimum\ Working\ stress = 731.33\ N/mm2$$

So as per the above analytical calculations we got maximum working stress of 821.45 N/mm^2 and minimum working stress of 731.33 N/mm^2. Now by using that working stress values we calculate the working load the chain link plate can carry by using the following formulae.

Inner link plate:

Maximum working load for inner chain link plate-

$$Maximum\ Working\ stress = \frac{Working\ Load}{Resisting\ Area}$$

$$Maximum\ Working\ Load = 269724.34\ N$$

Minimum working load for inner chain link plate-

$$Working\ stress = \frac{Minimum\ Working\ Load}{Resisting\ Area}$$

$$Minimum\ Working\ Load = 237938.95\ N$$

From above calculation we got the working load range i.e. varies from 2,69,724.34 N to 2,37,938.95 N for chain inner link plate.

Weight optimization functions

Objective functions define the objective of the optimization. An objective function is a single scalar value extracted from a design response, such as the maximum displacement or the maximum stress. An objective function can be formulated from multiple design responses. If the objective functions minimize or maximize the design responses specify, the Ansys Shape Optimization Module calculates the objective function by adding each of the values determined from the design responses. In addition, if multiple objective functions are used, a weighting factor is used to scale their influence on the optimization.

For example, if the design responses are defined from the strain energy of the nodes in a region, the objective function could minimize the sum of the design responses; i.e., minimize the sum of the strain energy, in effect maximizing the stiffness of the region.

An optimization problem can be stated as:

$$min\left(\Phi(U(x), x)\right)$$

$$\Phi_{min} = min\left(\sum_{i=1}^{N} W_i\left(\varphi_i - \varphi_i^{ref}\right)\right)$$

minimizing the maximum design response attempts to minimize the stress in the region that is exhibiting the maximum stress. The formula can be stated as:

$$\phi_{minmax} = min\left(max\left\{W_i\left(\varphi_i - \varphi_i^{ref}\right)\right\}\right)$$

WEIGHT OPTIMIZATION FOR INNER LINK PLATE:

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. The Fig. shows the model Concept-1.

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i.Concept 1:

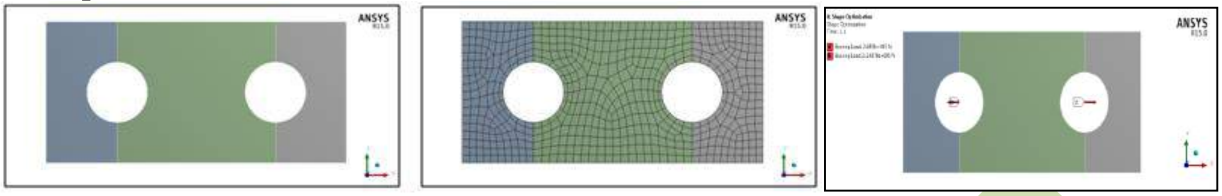

Fig No. 1. Geometry, Mesh model and Boundary conditions of Concept-1

Fig. 1 shows the Geometry and Mesh model of Concept-1. And Fig. 2 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.

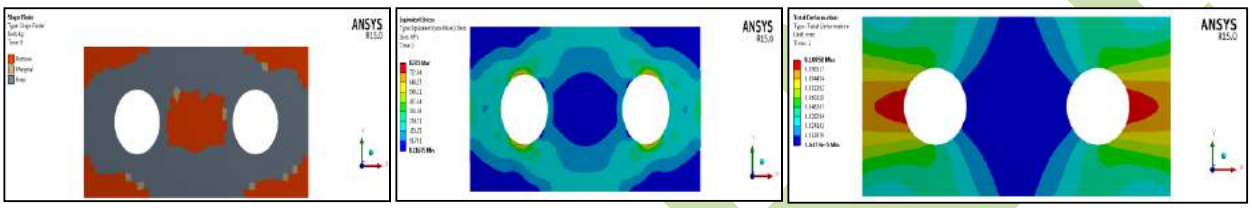

Fig. No.2. Shape Finder plot, Von Misses Stress plot and Displacement plot of Concept-1

Above Fig. 2 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. 1 and Shape Finder plot as shown in Fig. 2, 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.

ii. Concept 2:

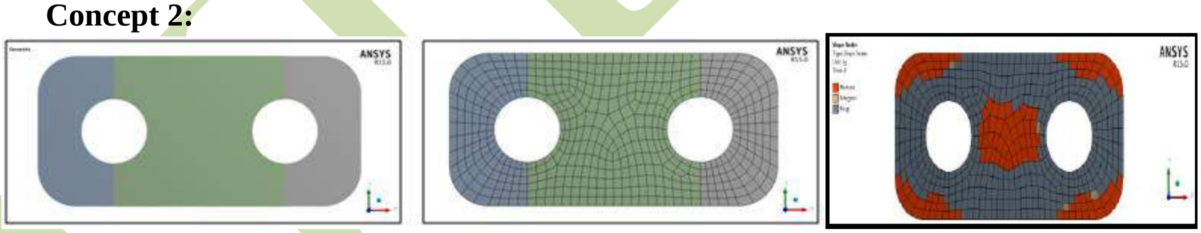

Fig No. 3. Geometry, Mesh model and Boundary conditions of Concept-2

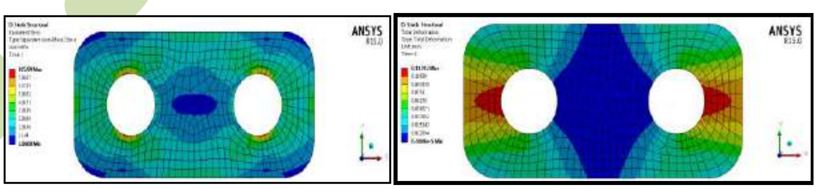

Fig. No.4. Shape Finder plot, Von Misses Stress plot of Concept-2

From Equivalent Stress plot and Shape Finder plots of this iteration as shown in Fig.3 and Fig. 4 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.

iii. Concept 3:

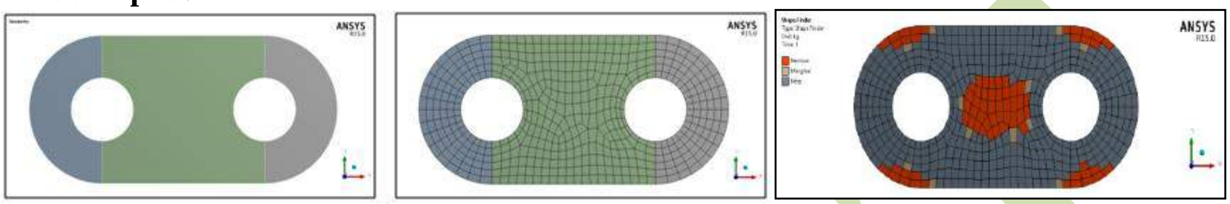

Fig No.5. Geometry, Mesh model and Boundary conditions of Concept-3

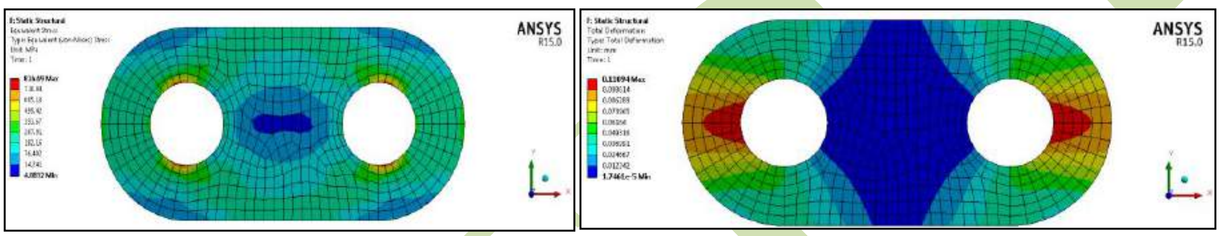

Fig. No.6. Shape Finder plot and Von Misses Stress plot of Concept-3

In Concept-3 model is came to half circular ends for the link plate by following previous Shape Finder and Von Misses Stress plots contours. From the Concept-3 Shape Finder and Von Misses Stress plots contours the optimum shape for the ends of the link plates. From Concept-3 Von Misses stress and Shape Finder plot it is observed the scope of reducing the material and changing the shape of the link plate at the centre of the plate. It is decided to follow the shape contour of the centre shape finder plot and reached to the oval shape hole at the centre.

Concept 4:

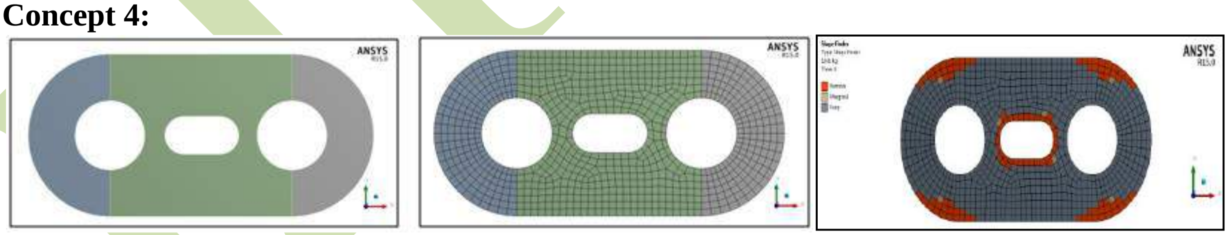

Fig No.7. Geometry, Mesh model and Boundary conditions of Concept-4

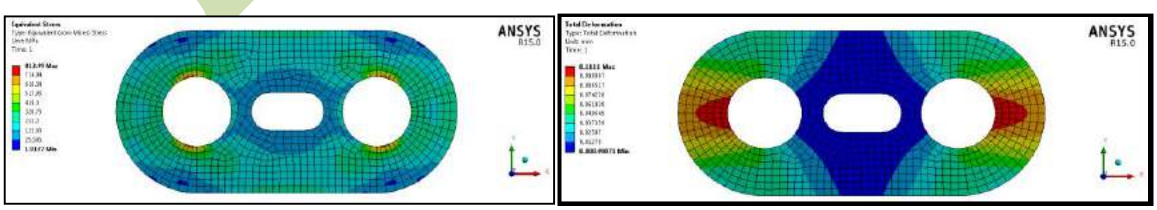

Fig. No.8. Shape Finder plot and Von Misses Stress plot of Concept-4

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In Concept-4 Von Misses Stress plot and Shape Finder plot it is observed there is still scope for the material removal at the centre portion of the link plate and hole size is increased by following the shape finder contour from the shape optimizer to get to the next iteration design.

iv.Concept 5:

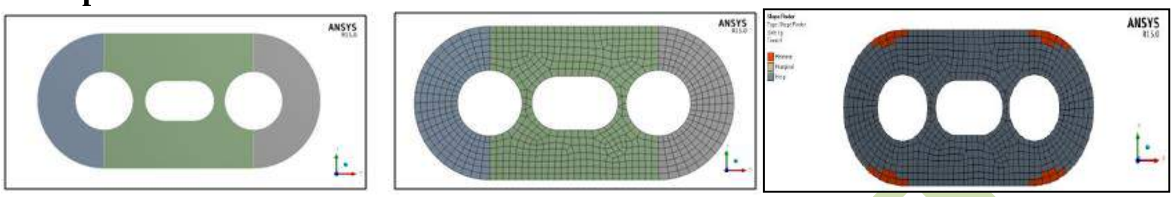

Fig No.9. Geometry, Mesh model and Boundary conditions of Concept-5

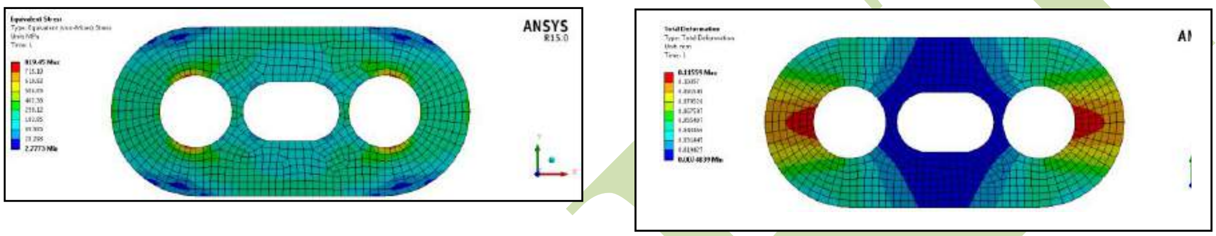

Fig No.10. Shape Finder plot and Von Misses Stress plot of Concept-5

In Concept-5 Shape Finder plot from the shape optimizer module any portion of the link plate in the removal zone is not observed as per suggestion by the ANSYS solver. But following the general design used in the industry and low deformation region observed at the central edges of the link it is derived to the next iteration of the link plate design by removing the material from the same region.

v.Concept 6:

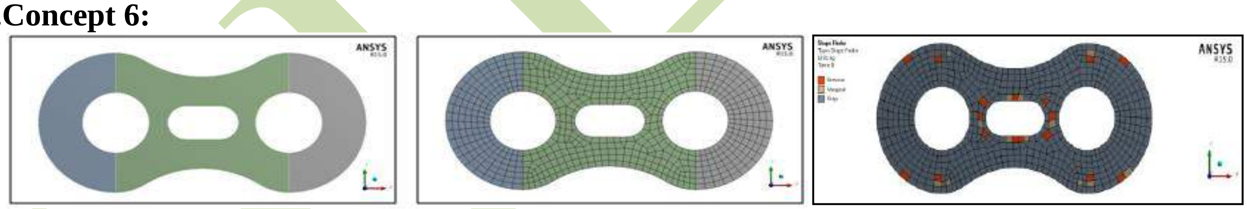

Fig No.11. Geometry, Mesh model and Boundary conditions of Concept-6

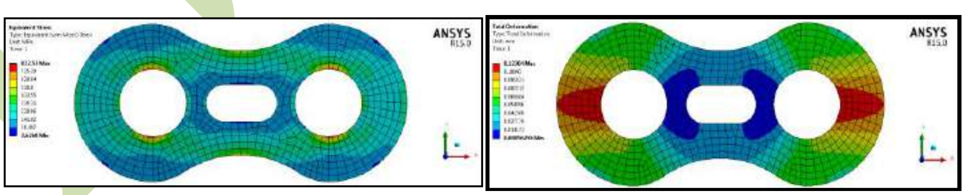

Fig No.12. Shape Finder plot and Von Misses Stress plot of Concept-6

In Concept-6 it is observed that stress in the central edge region still more material can be removed and next iteration is derived. The middle slot increases deformation of link plate. As link plate used is meant to be used for load transmitter it needs to be rigid and needs additional rigidity at its peak load at which high stress concentration near hole vicinity may occur. Central slot is removed in next iteration.

Concept 7:

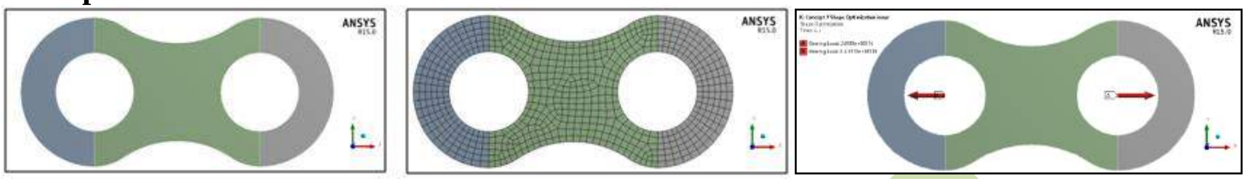

Fig No.13. Geometry, Mesh model and Boundary conditions of Concept-6

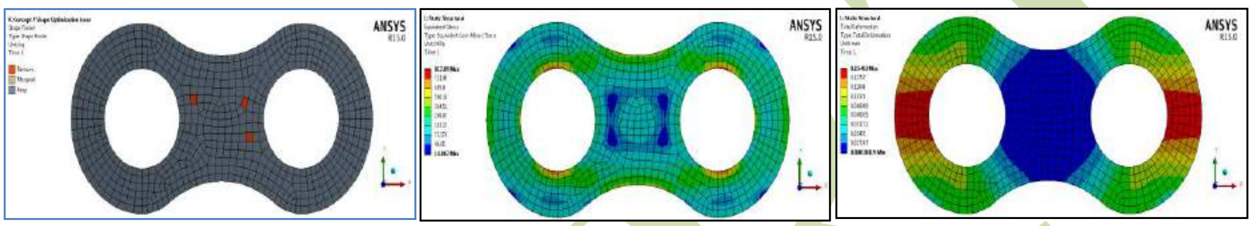

Fig. No.12. Shape Finder plot, Von Misses Stress plot and Displacement plot of Concept-7

Central slot is removed in above iteration as link is kinematic mechanism which must be closed and should only contain slots where coupler curve is required. As link plate used is meant to be used for load transmitter it needs to be rigid and needs additional rigidity at its peak load at which high stress concentration near hole vicinity may occur. Final iteration above shows evolved model through history followed by engineering statistics to gain required link model as per standard topology optimization process.

Experimental testing of Roller Chain Link Plates:

Fig. No. 13. Experimental set up

For experimental testing of roller chain link plates as shown in Fig. 13 link plate and fixture fixed in UTM Machine. A Computerized Universal Testing Machine of 60 Tons capacity is used. It is made by Fine Tune Engineers Pvt. Ltd. Miraj, Sangali. This testing is done at Material Testing Laboratory of Akhil Bhartiya Maratha Shikshan Parishad's Anantrao Pawar College of Engineering & Research, Parvati, Pune. Roller chain's links are clamped between the two jaws of UTM by using fixture. In UTM two heads are shown upper one is movable and another one is middle head which is fixed.

Fig. 14 Chain Links before testing

Fig. No. 15. Roller Chain Links after testing

In experimental test result it shows that the failure has taken place in the link plate means where cross section area is minimum. In Fig. No. 14 shows that one link assembly made by using conventional manufacturing (press tool) method and another roller chain link assembly is made by using non conventional manufacturing (EDM Machine) method. Also his picture shows both roller chain link assemblies before test or before break. And fig. No. 14 shows that both roller chain picture after testing or after break. The result shows that both roller chain link assemblies break same location which is outer link plate near to hole where cross section area is minimum.

But the breaking load is different. By using conventional manufacturing process roller chain link assembly breaks at 56.51 Tons and anther one manufacturing by using conventional manufacturing process roller chain link assembly breaks at 62.83 Tons. Hear clearly shows the uncertainty in manufacturing which directly effects on breaking load or strength of roller chain link assembly.

Theoretical, Finite Element Analysis and Experimental Test Results:

Theoretical calculation of allowable maximum stress have been carried out for Outer and Inner link plate, pin, bush and roller chain link assembly. Also Finite element analysis has been carried for both outer and inner chain link plate, pin and the roller chain link assembly and experimental work also carried out for only both outer and inner link plates and roller chain link assembly. Von misses stress and displacement has been obtained for different components of roller chain link assembly. Below table gives the comparison results for theoretical, numerical (Finite Element Analysis) and Experimental test for both outer and inner link plate, pin, bush and the roller chain link assembly under the static tensile loading condition.

Table No.1 Results of Chain link plate testing

Sr. No.	Sample Code	Material	Von Misses Stress (N/mm²)	Theoretical Stress (N/mm²)	Experimental Breaking Load (kN)
01	Z1	EN-8	817.09	821.45	2,74,275
02	Z2	EN-8	817.09	821.45	288.600
03	Y1	EN-9	817.09	821.45	296.280
04	Y2	EN-9	817.09	821.45	299.370
05	X1	EN-19	817.09	821.45	327.900
06	X2	EN-19	817.09	821.45	289.740

The above Table 1 shows theoretical stress of inner link plate is 821.45 N/mm² and as per numerical it is coming 817.09 N/mm² respectively and it is less than allowable limit, so the chain link plate design is safe. Also its breaking load is 2,74.275 kN, 288.600 kN, 296.280 kN, 299.370 kN, 327.900 kN and 289.740 kN respectively.

As we seen the above result of analytical, experimental and numerical behavior of link plates under tensile loading. The fatigue initially nucleated at the external cracks of the chain link, and later propagated to the inside of the links until sudden fracture occurred. As the Finite element analysis results are within +/- 10% of the calculated working stress, so the chain link plate, pin, bush were safe under the maximum working load of 25 Tons. As we designed and developed a chain as per power transmission capacity, it was observed experimentally that the roller chain links failed at 274.275KN, 288.600KN, 296.280KN, 299.370KN, 327.900KN, 289.740KN. which are above the designed value.

CONCLUSIONS

It is concluded from the survey of many industries, the major modes of roller chains occur in their link plates. The experimental analysis has demonstrated the effect of material and manufacturing uncertainty by taking example of 6 chain link plates. It has been observed that there was high stress concentration near to hole in the chain link plate therefore breaking takes place at the location of minimum cross section area. Based on the FEA and Experimental results, it is observed that the optimal value of radius is between 44.5 to 45 mm. The weight saving thus achieved will have a significant impact on cost of the chain, and more importantly with a lighter chain, the cost savings during operation will also be significant. As the Finite element analysis results are within +/- 10% of the calculated working stress, so the chain link plate, pin and assembly were safe under the maximum working load of 25 Tons. As we designed and developed a chain link as per power transmission capacity, it was observed experimentally that the roller chain links failed at a load more than 29.50 Tons.

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