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Tushar D. Bhoite
Department of Mechanical Engineering, SVERI's College of Engineering, Pandharpur, Solapur, Maharashtra, India, tus_bhoite@rediffmail.com

Prashant M. Pawar
Department of Mechanical Engineering, SVERI's College of Engineering, Pandharpur, Solapur, Maharashtra, India, PrashantM.Pawar@gmail.com

Bhaskar D. Gaikwad
Department of Mechanical Engineering, SVERI's College of Engineering, Pandharpur, Solapur, Maharashtra, India, BhaskarD.Gaikwad@gmail.com

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Fea Based Study of Effect of Radial Variation of Outer Link in A Typical Roller Chain Link Assembly

Tushar D. Bhoite, Prashant M. Pawar & Bhaskar D. Gaikwad

Department of Mechanical Engineering, SVERI's College of Engineering, Pandharpur, Solapur, Maharashtra, India
E-mail : tus_bhoite@rediffmail.com

***Abstract* -** Chain Link assembly is extensively used in the industry, the scope of this paper is to review the applications in the industry and explore the design considerations that go into the design of the assembly.

The paper delves into various application aspects and manufacturing aspects to formulate an idea of the system.

Finally Finite Element Analysis (FEA) has been used to conduct shape optimization. Since lot of work has already been done in other components, in this paper the focus has been narrowed down to specific component of outer link.

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. In this paper we assess the impact of this radius on the stress in the system, and see if material saving and consequently efficiency increment is possible.

***Key words* -** *roller chain, link plate, FEM analysis, stress analysis, shape optimization.*

I. INTRODUCTION

Economy of state is dominated by agricultural as well as industrial sector. Sugar factories play important role in economy of state. About 60 percent processes in these factories are based on roller chain conveyers. Apart from that, other industries also use these chains frequently for process atomization. However, failure of this chain is perennial problem in these industries which causes huge losses to these industries along with its dependants and in turn economical growth of the state. So, roller chain is the most important element of the industrial processes.

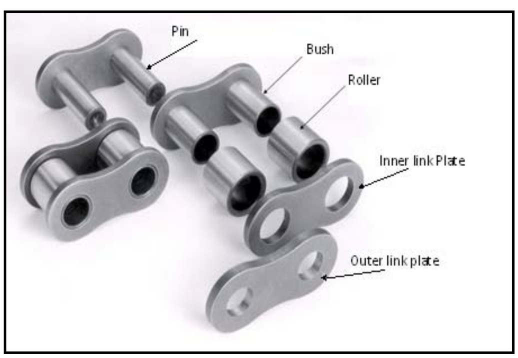

Fig. 1: Shows the typical roller chain link assembly

Most of the time chain is under tension which causes failure of chain assembly which is the major problem for industrial sector. Causes of this failure are improper design. It is important to study the influence of these parameters. All these parameters can be considered simultaneously and chain link design optimally. 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 shape optimization process is used for the design of roller chain link for minimization of failure modes. This process various design variables, such as wall thickness of link, breaking area of link and shape of the link. While deciding the shape optimization of 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 EN8, EN9 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 mm^2.

Two types of roller chains are in common use: single strand and multiple strands. In multiple strand, two or more chains are assembled side by side on common pins that maintain the alignment of the rollers in the several strands. Standard roller chains are defined as *pitch proportional*, which makes them different from other types of chains with rollers. The ASME standards' nominal dimensions for these chains areapproximately proportional to the chain pitch. The pitch of a roller chain is the distance between the centers of adjacent joint members.

Some innovations were devoted on improving chain components and articulation of the chain. Hollingworth and Hills [2] worked on forces in a heavy-duty drive chain during articulation. A rigid-body analysis is presented of the forces which occur in chain bearing in an articulation. The analysis differentiates between the two types of chain bearing open end leading or open end trailing and the results show that the force characteristics in each are significantly different. Experimental verification of the chain link tension is good. Miyazawa and Satoh [3] developed a method of manufacturing a link plate for a roller chain which results in minimization of the link plate deformation, a bending failure generated by the interference between a warped link plate and the adjacent link plate. Moster and Ledvina [4] developed a roller chain assembly by adding material to the location on the link plate face where fatigue failure is most likely to occur. The material increases the strength of the link plate, allowing for the use of inner links having larger bushings.

Apart from patents on chains and conveyer, few researchers have published theoretical and experimental works in this area. Initial research on chain [5, 6] was focused on kinematics and dynamics study of chain and sprocket combination. An analysis to determine the effects of pitch difference, friction, and centrifugal forces on the load distribution of a roller chain was carried out by Naji and Marshek [7]. Friction was shown to cause higher tooth load on driven sprocket than that of a driver sprocket. This research area was further extended for dynamic analysis of oscillation in chain drive and computerized roller chain drive selection [8-9].

Very few researchers worked on stress analysis of roller chains. Ozes and Demirsoy [10] examined the effects of various loading condition on the stress of a pin-loaded woven-glass fiber rain forced epoxy laminated conveying chain component. A numerical & experimental study was carried out to determine the stress distribution of composite conveying chain components used to convey loads. The commercial finite element package ANSYS was used to perform the numerical analysis using a three dimensional eight-nodded layered structural solid elements. Chain tensile forces were loaded through pins and chosen as 250, 500, 750, 1000 and 1250 N for the two conditions of chain components. Experimental and numerical studies were compared and discussed for two conditions and five different tensile forces. A good agreement between experimental results and numerical result predictions was obtained. [11] Design team worked on the ANSY size 60 roller chain is sufficiently sized to safety operate when connected to the 10 hp engine used by the mini Baja. This size roller chain is typically used in motorcycles with engines capable of producing 50 to 70 hp. The Finite Element study conducted here estimations of the design team. Noguchi et al [12] proposed some methods of weight saving for roller chains. These methods are based on Finite Element Method analysis of the stress and deformation in the link plate of roller chain and also approaches for reducing stresses and weight saving in the link plate of the roller chain. Stress are 3% higher in the proposed design, but the weight is reduced by 10%. Tensile tests are performed on link plates made of resin, and the effectiveness of the proposed model is confirmed. Korey et al [13] two dimensional geometrical model of the chain link is formed and stress analysis is performed using both boundary element and finite element methods. The researchers are proved Boundary Element Method is more appropriate for the chain link application than the finite element method.

After the analysis work very few researchers were developed on optimization of roller chain. Burgess and Lodge [14] investigate the optimization of the chain drive system on sports motorcycles. The transmission efficiency of the chain drive has a very significant effect on the performance of sports motorcycle. Sergeev and Moskalev [15] have worked in investigating the coupling in a chain transmission. This entails investigation of the coupling geometry and parametric optimization of the sprocket. Dasari and Ramesh [16] investigate the analysis of a complex shape chain plate using transmission by using photoelastic technique. Here researchers explained how transmission photoelastic technique is used to estimate the stress distribution and its concentration zones in a complex chain late when it isolated.

From the various studies, it can be noted that, even though several patents are filed on roller chains, most of the patents based on improvement of efficiency and performance. Hardly here are very few patents available which focuses on improving life of the chain and minimization of its failure. It can also be noted that the analytical work in the literature is focused on load estimation. Very few researchers have explored the fatigue life estimation and stress analysis for the chain assembly. However, literature on uncertainty analysis due to improper shape of roller chain is present. The failure case studies also indicate that the birth of some failure modes is given at the time of designing stage itself.

II. LOAD CONSIDERATIONS

A. Tensile Load (Nominal Tensile Load)

The main consideration for all types of chain is the nominal tensile load that is required to perform the basic function. The nominal tensile load generally fluctuates in a regular cycle. Figure-2 roughly shows how the tension varies in a chain that is 100 pitches long as it runs around 20-tooth sprockets. This nominal tensile load is the basic load considered in almost all chain ratings.

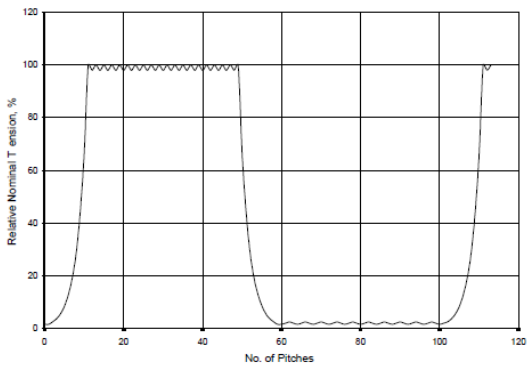

Fig. 2 : Tension varies in a chain

B. Shock Load

Shock loads are caused by the characteristics of the power source and the driven machinery. They occur repeatedly in a regular cycle, usually one or more times in each shaft revolution.

C. Inertia Load

As the term is used here, inertia loads are different from shock loads. Inertia loads are the occasional loads imposed on the chain by unusual, and often unexpected, events. They may come from starting a heavily loaded conveyor or a drive with a large flywheel. Or they may be caused by a sudden momentary jam in the driven machine or conveyor.

D. Centrifugal Tension

In high-speed drives, centrifugal force is generated as the chain travels around the sprockets. Centrifugal force also may be generated by the chain's travel over a curved path between sprockets.

E. Catenary Tension

The weight of that portion of the chain that hangs in a catenary generates additional tensile loads in the chain.

F. Chordal Action

As the chain wraps a sprocket, it effectively forms a regular polygon. That causes the chain strand to rise and fall each time a joint engages a sprocket tooth. This motion is called chordal action, and the effect is illustrated in Figure-3. The tension in the chain changes slightly every time the chain speed changes.

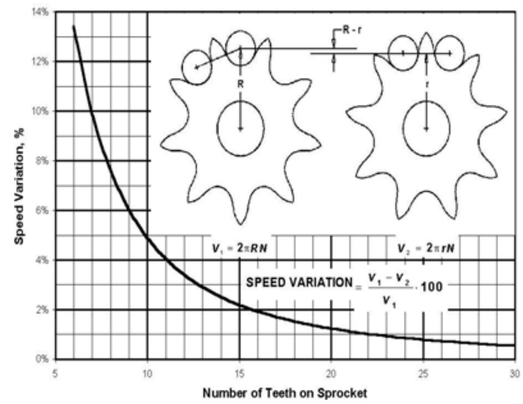

Fig. 3 : Chordal action

G. Vibration

Chain vibration can cause very large increases in chain tensile loading if the vibration occurs at or near the natural frequency of the chain.

III. DESIGN CONSIDERATIONS

Roller chains are used in a wide variety of applications, but most roller chain is used in drives. The shaft speeds of the drives range from less than 50 rpm to nearly 10,000 rpm, and the amount of power transmitted ranges from a fraction of a horsepower to more than 1000 hp. The main design considerations for a roller chain to be used on a drive are the various tensile loads.

A. Ultimate Tensile Strength

The ultimate tensile strength of a chain is the highest load that the chain can withstand in a single application before breaking. It is not a major consideration in designing roller chains. It is only important because yield strength and fatigue strength depend on ultimate tensile strength. Minimum ultimate tensile strength (MUTS) is a requirement in the ASME standards that govern roller chains. A well-made roller chain almost always meets the standard.

B. Yield Strength

The yield strength of a chain is the maximum load from which the chain will return to its original state (length). For many standard chains, the yield strength is approximately 40% to 60% of the minimum ultimate tensile strength.

Figure-3 shows a typical load elongation diagram for chain. The figure clearly shows that the yield point for the particular chain shown is at 60% of the ultimate tensile strength. Yield strength is an important consideration in designing roller chains. For standard roller chains, conforming to ASME, the yield strength is about 60% of the MUTS. Figure-4 is a diagram of how a standard roller chain elongates as a tensile load is applied.

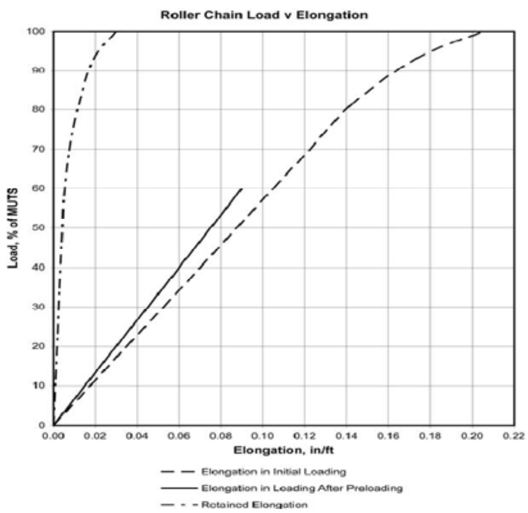

Fig. 4 : Standard roller chain elongates as a tensile load is applied

IV. FINITE ELEMENT ANALYSIS

FEA modeling consisting of modeling of chain link, preprocessing, processing and post processing in ANSYS Workbench 12.0

Following Figure-5 shows that 2D CAD model of chain link with dimensions.

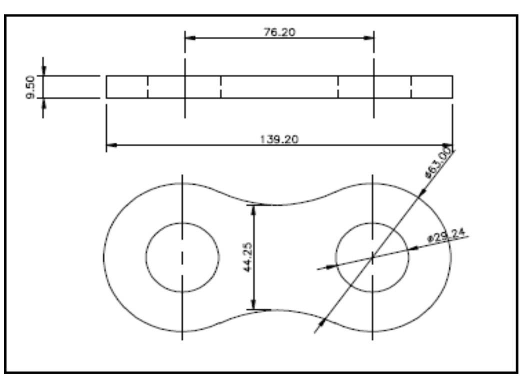

Fig. 5 : Shows the 2D CAD model

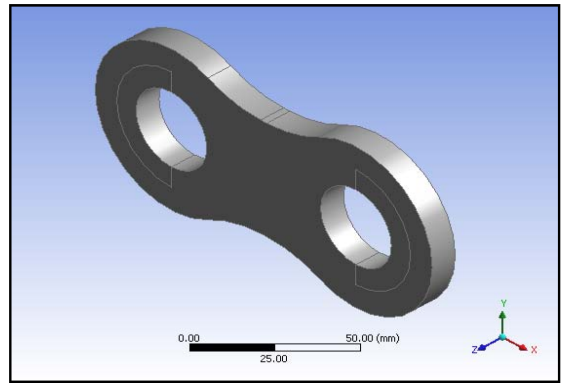

Fig. 6 : Shows the chain link modeled in ANSYS Workbench 12.0

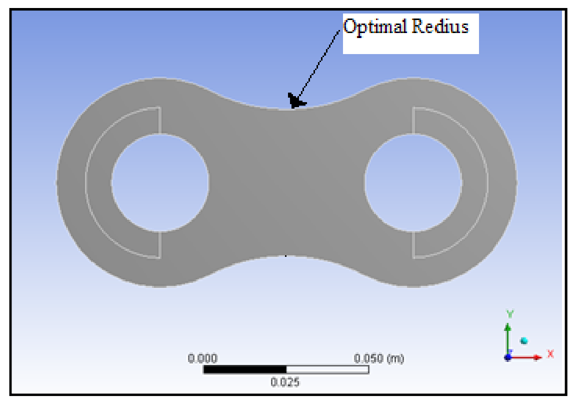

Fig. 7 : Shows the optimizing radial distance

After modeling the chain link meshing is done in ANSYS Workbench. Meshing involves converting of geometry into nodes and elements.

3D Hex Dominant mesh type is used for meshing the 3D model in ANSYS Workbench 12.0 .Single entity set named as Mesh body consisting of total solid body is created. Second order Solid95 element type is used for analysis. Three different element sizes are used as 1mm, 2.5mm and 5mm for refinement for checking divergence.

TABLE I. MESH SIZE WITH APPROXIMATE NO. OF ELEMENTS

Sr. No.	*Mesh Size (mm)*	*Approximate No. of elements*
1	1 mm	67934
2	2.5 mm	5232
3	5 mm	1250

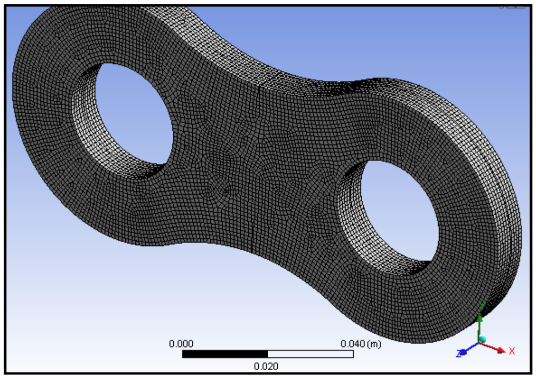

Fig. 8 : Shows the meshed model with 1mm element size

Solid95 is higher order version consisting 20 node[17]. It can tolerate irregular shapes without as much loss of accuracy. Solid95 has compatible displacement shapes and are well suited to model curved boundaries. The element is defined by 20 nodes having three degrees of freedom per node that is translation in nodal X, Y, Z directions. The element may have any spatial orientation.

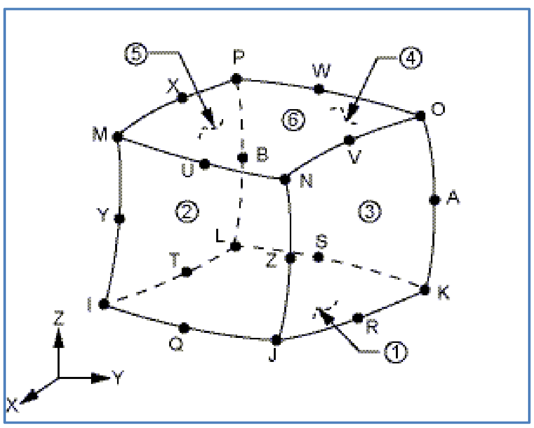

Fig. 8 : Solid 95 Geometry

While modeling link, imprint faces are created which are useful for selecting particular faces at the time of applying boundary condition (figure-9)

As shown in figure-9 the blue face indicates the fixed support and tensile force of 100 N is applied to red face in X direcection. By varying the optimal radius as shown in fig from 45mm to 55mm with the 0.5mm incremental steps the results are analyzed in three different element size viz 1mm, 2.5mm, 5mm and number of elements in each case are noted.

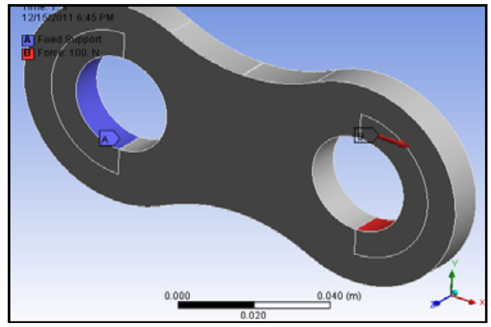

Fig. 9 : Applied boundary conditions

The FEA results of link for deformation and stress using 1mm element size are shown in figure-10

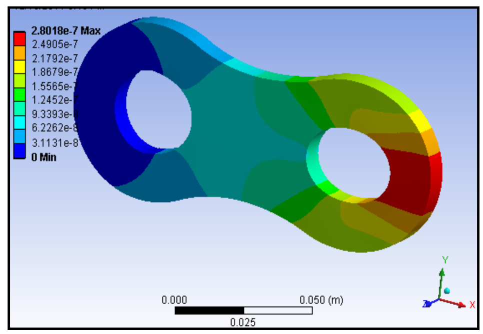

Fig. 10 : Deformation in link

As shown in above fig maximum deformation of 2.81mm is occurs at the region where the force is applied.

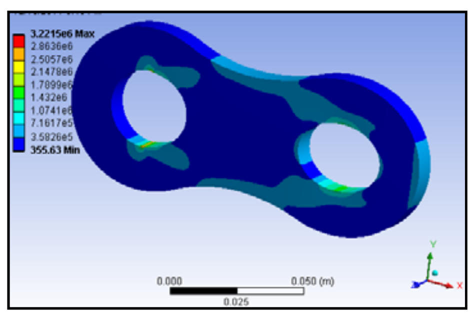

Fig. 11 : Stress distribution in link

The region around two circles shows stress concentration of 3.22MPa.

The graph shown in figure-12 is stress vs radius for three different element sizes. It is clear that as element size decreases stress goes on increasing and the stress difference between element size 2.5mm and 1mm shows the case of divergence showing stress concentration.

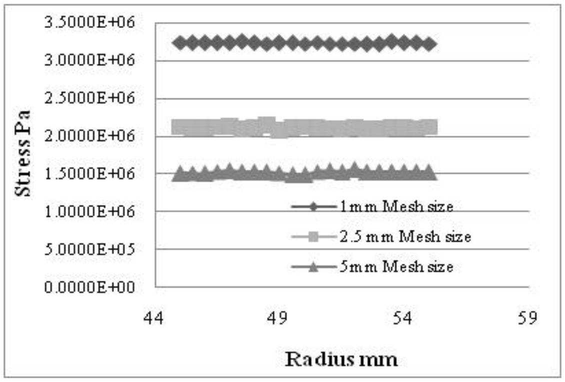

Fig. 12 : Stress distribution of Link

Since we are concerned with optimization, instead of resolving stress concentration, it is assumed that for a particular mesh size the error will be approximately same and hence we shall focus on stress variation for a single mesh size.Figure-13 shows the graph of stress vs radius for 5mm mesh size. The lower value of stress 1.485MPa at the radius 50mm shows the optimal result.

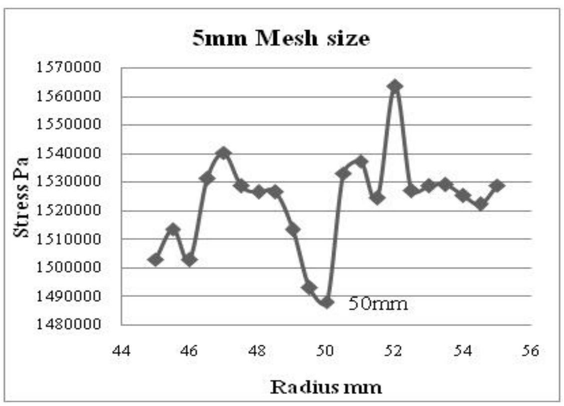

Fig.13 : Optimal value of Radius

V. CONCLUSION

Based on the FEA results, it is observed that the optimal value of radius is between 49.5 to 50 mm. Though this optimization seems insignificant on its own, it must be noted that in a typical industrial application, thousands of such links will be needed. 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.

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