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Design and Development of Curve Shape Chain for Transmission Drive Unit of Rotary Cage

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ABSTRACT

Horizontal indexing rotary cage holder is a part of the multi indexing machine. Inside the multi indexing machine the operations such as facing, machining, drilling and air blow cleaning are usually done on cylinder block. In the current process manual adjustment is done by an operator which consumes 25% of the cycle time. The aim of the present work is to design a unique drive system to rotate the cage and reduce the cycle time. It can be carried out by using gear drives and chain sprocket drives. As the diameter of the cage is large thus gear drives are costlier. The ring provided for cage body is bent from sheet metal. Thus new curved chain link is to be developed in its shape and dedicated size. The appliedtorque and forces to the drive are calculated by the standard analytical procedure. The modeling is done on CATIA software and analyzed on ANSYS Workbench. Validation is done by comparing analytical and experimental results taken from UTM machine.

Keywords

Chain drive; Sprocket; Rotary cage; Indexing; Gear drive

INTRODUCTION

Power can be transmitted by various drives such as belt drive, chain drive and gear drive. The selection depends upon space restriction. For Large distance rope, chain and belt are used. For small distances gears are used. Power transmission can be effectively done by using a chain drive. All these drives have some advantages and disadvantages. If the considered centre to centre distance then distance has limitation in case of gear. For a given set of gear there are specific dimensions. But in the case of chain center distance between two chained sprockets can vary from 50% to 300% or more of their pitch diameter. Considering the installation factor chain drives are very easy to install. During assembly also chain drives are not restrictive as those for gear drive. Design and reconfiguration is easy in comparison with gear drive system. During shock loading condition chain drive performs better than gear drive. In case of chain drive operating load gets distributed over many teeth where as operating load acting on gear drive is concentrated on one or two teeth. But bearing load gets reduced as chain drive not requires tension in slack side. Working with minimum of machine tool chain drives are better options. Here the objective of present work is to develop new curved chain connections with its strength validation and the unique design for its shape and dedicated size.

LITERATURE REVIEW

Kai Wang, et al. [1] investigated that the die clearance has great impact on resulting edge fracture and developed computational methodology for edge fracture prediction and carried out FE analysis of hole punching process.

Tushar D. Bhoite, et al. [2] found that failure modes can be reduced by doing shape optimization of roller chain link.These processes various design parameters such as wall thickness, breaking area and shape of the link. By doing FEA analysis, optimum radius valve is decided. Though it seems to be small change, butthe large quantity of chains are required for application thus weight saving is more.

James C. Conwell, et al. [3] discussed dynamic behavior of roller chain. The strain gauges are mounted on chain is used to find chain tension during normal load application. Wide range of liner chain speed and preloads are also considered. During testing special idler sprocket is used which gives a measurement of horizontal and vertical components of the bearing reaction force. Tension in chains increased while going from loose side to tight side and vice versa.

Pusit Mitsomwang, et al. [4] concluded that punch or die sharing is one of important method. Cutting characteristic get affected by punch and die clearance. The crack propagation is studied and analyzed effect of feed velocity.

Wang Hong [5] studied the importance of fine blanking and stated that quality of fine blanking parts is directly related to quality of mould design. Analytical calculation of layout design, fine blanking pressure and force are done.

Masao Murakawa [6] introduced the new technology to improve surface quality of sheared product. For this two processes finish blanking and press shaving are combined.By carrying out the process, small clearance between punch and die is found due to the processes like tempering, quenching, grinding and turning are eliminated.

M. D. Jagtap, et al. [7]carried out the experimental demonstration of strain gauge on chain strip in a Strain Rosette - 45˚ manner. Finally, experimentation is carried out on Computerized Universal Testing Machine (UTM).

Nur Ismalina BintiHaris [8] studied the cause of failure of the chain system by characterization on the failed component. Weld defect cause crack propagation and cyclic loading cause fatigue failure. Fatigue failure occurs within chain attachment and outer chain link plate. The scanning electron microscopy revealed the types of microstructure at the heat affected zone. Rockwell tests found a different hardness profile at three areas weld metal, base metal and heat affected zone.

Ridha Hambli [9] discussed the importance of ring indenter in fine blanking process and done the numerical simulation of fine blanking operation and states the importance of fine blanking process.

G. Pantazopoulos, et al. [10] discussed that the chain link which is used in drawing benches is subjected to repetitive tensile stresses during loading and unloading. Friction caused due to improper lubrication can be very harmful, failure of chain link causes increased machine down time resulting in poor productivity. The fatigue mechanism starts at inner eye area followed by brittle overloading fracture.

P. Sadagopan, et al. [11] analyzed wear reduction of existing chain used in 100cc motorcycle. Comparison is done between field results and calculated elongation. The new design is developed and theoretical evaluation is done and compared with existing design. Fatigue properties are evaluated based on mathematical model and by ANSYS software.

R.K. Pathak [12] studied die clearance and found out expression of die clearance up to proportion limit. There is need to find alternate manufacturing method of metal meso-scale that can compete and outperform silicon-based lithography processing for making elements in sensor and actuators.

S. Noguchi [13] studied the experimental testing of roller chain. For doing weight saving different design solutions are provided. Depend on static stress analysis roller chain weight saving is done with suppression of increase in stress by FEA software. In FEA different loading conditions are considered and stress is considered as benchmark for weight saving. ABS resin is used for making prototype, under different loading condition and verified by tension test.

V.Kerremans, et al. [14] studied the wear of roller chains and found that there is no reliable test-ring to measure the chain wear. During these different components of roller chain and loading conditions are described. Polymer chains disadvantages are discussed. Wear between pin, bushing and roller are considered.

Alan James [15] studied three chain systems used widely in different applications. Different operating conditions of chain system include a furnace hearth, a drilling rig for rock and finally dragline crane used for removing overburden.

EXISTING SYSTEM

Fig. 1 shows the schematic of existing system. Automated rotary cage type concept for cylinder block is taken from the special purpose machine in which the component of the cylinder block is to be drilled, cleaned, and proceeds towards assembly section in continuous production line of automobile company. The arrangement of process wise well defined sequence and operation for the decided cycle time. Where the rotary fixture along with component will get stoppage at every angular position with the used sensors. Operation cycle will run through PLC Programmed.

Fig. 1 Manual rotation and locking without full proofing

DESIGN AND ANALYSIS

A. Material selection

Selecting the right material for parts of drive many factors are considered, such as cost and material performance. Generally steel is used as material, which is alloyed of iron and carbon but stainless steel is an alloy of iron, chromium and nickel. In alloy steels major component is carbon and other elements are added to increase hardness. When the alloy is made, heat treatment ofthe material becomes easy. Mostly chromium, nickel and molybdenum are added. Generally SS304 (for medium strength, medium corrosion resistance), SS316 (For low strength, high corrosion resistance), SS600 (For high strength, low corrosion resistance). Here the application requirement is anticorrosive properties and medium. So that SS304 is selected.The ring provided to the cage body is bent from the sheet metal part.

B. Modeling of Drive

Fig.2 shows the final design of curved shape link. For finalization of shape here the five iterations are consideredwith the different inner and outer radius. Here different shapes has been carried out with higher strength and deformation analysis also carried out and finally choosen the dimension.

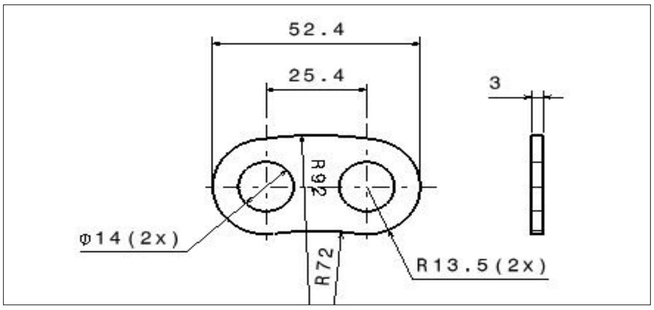

Fig. 2 2D Drawing of chain linkage

C. Stress calculations for chain linkage

- **For curved chain**

Here,

Uniaxial tension parallel to row of holes

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Tension = 520 N
Diameter of hole, d = 14 mm
Width of plate = 20 mm
Thickness of Plate, t = 4 mm
Maximum tension stress, $\sigma_{max} = \sigma_{nom} x\ K_t$
Where, σ_{nom}= Nominal stress= normal stress
Hence Nominal Stress, $\sigma_{nom} = P / A$
$\sigma_{nom} = 8.67\ N/mm^2$
and
K_t = Stress concentration factor
$K_t = 3.000 – 0.712\ (d/L) + 0.271\ (d/L)^2$
$K_t = 2.69$
Hence
Maximum stress, $\sigma_{max} = \sigma_{nom} x\ K_t$
$\sigma_{max} = 6.5 \times 2.69$
$\sigma_{max} = 23.31\ N/mm^2$

Comparing these analytical results with finite element analysis. Consider load 1040N is divided on two sides of linkage. So total 520N load is applied on each side. Select static analysis model and select material SS304 in engineering data and mesh the link and applied boundary conditions.

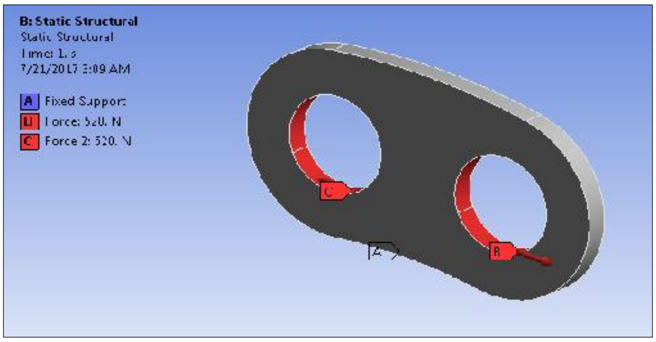

Fig. 3 Boundary Conditions Applied On Chain Linkage

Fig. 3 shows the boundary conditions are applied and after Appling boundary conditions the von-mises stress and deformation is evaluated.

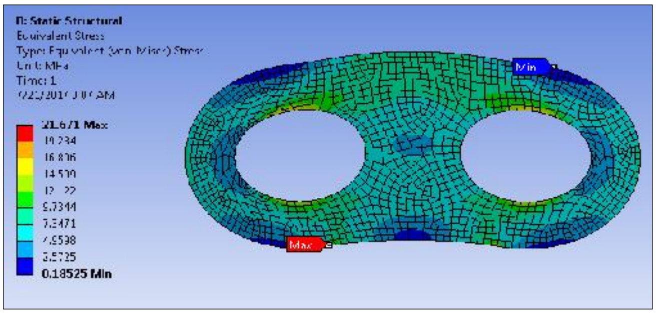

Fig. 4 Von-Mises stress results for chain linkage

Fig. 4 shows the maximum Von-Mises stress which is 21.671 MPa and in safe limit. The red colour shows the area with maximum stress and the region with blue color shows the area with minimum stress.

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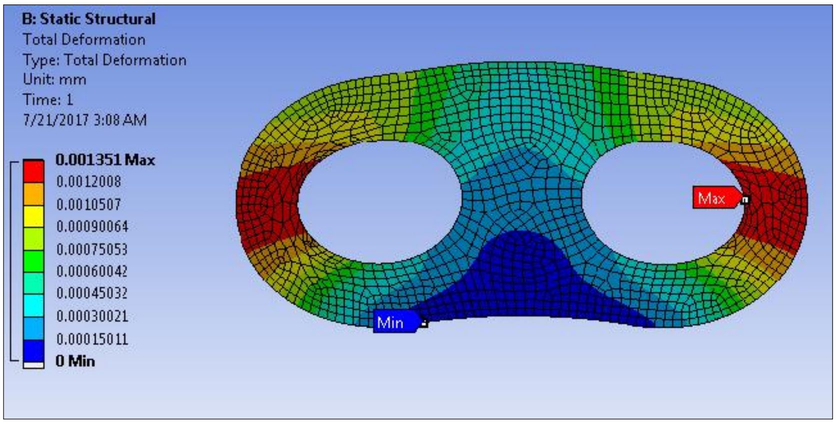

Fig. 5 Negligible deformation for chain linkage

Fig. 5 shows the deformation of curved shape link, the maximum deformation is 0.001351 mm which is negligible and in safe region.

- **For Pin**

Bending failure of pin

When the pin is tight, failure occurs due to shear. On the other hand, when pin is loose it is subjected to bending moment as shown in Fig. 6.

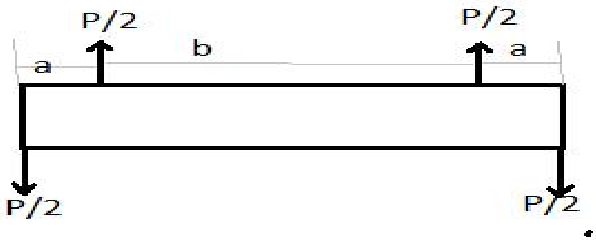

Fig. 6 Loading Diagram for Pin

Where, b =25mm

a = 4 mm

Now,

$\sigma_{b}= M_{b}y/I$

Where I = (π/64) d4

y = d/2

$M_b = (P/2) [(b/4) + (a/3)]$

Hence

$\sigma_b = 5.85$ N/mm^2

Normal stress

$\sigma_{nom} = P / A$

A=2 πr=2x π x9.5=59.69 mm^2

σ_n =8.711 N/mm^2

Pin is subjected to 520N on both sides the bending stress is a major area of failure so bending and Von-Mises Stress is calculated analytically and with the use of FEA software and the results are compared.Fig. 7 shows the boundary conditions for the pin.

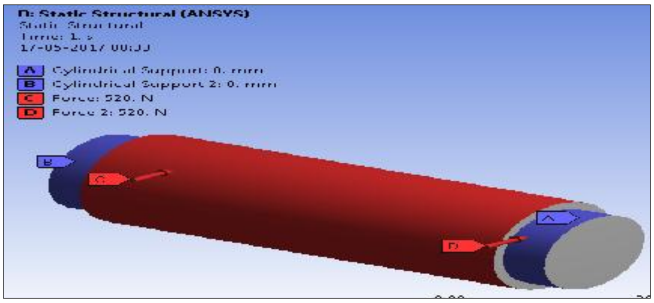

Fig. 7 Boundary condition for Pin

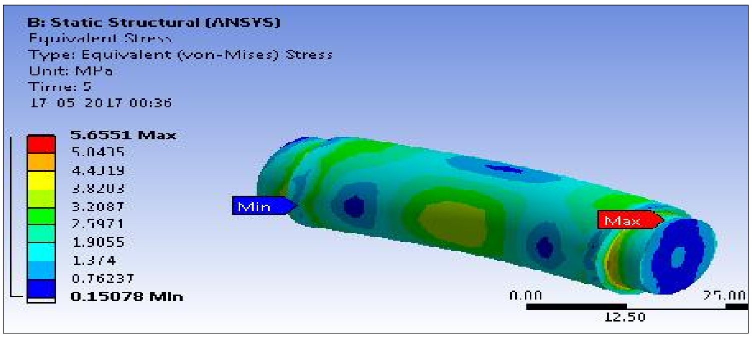

Fig. 8 Von-Mises Stress Results for Pin

Fig. 8 shows the maximum Von-Mises stress which is 5.6551 MPa and in safe limit. The red color shows the area with maximum stress and the region with blue color shows the area with minimum stress.

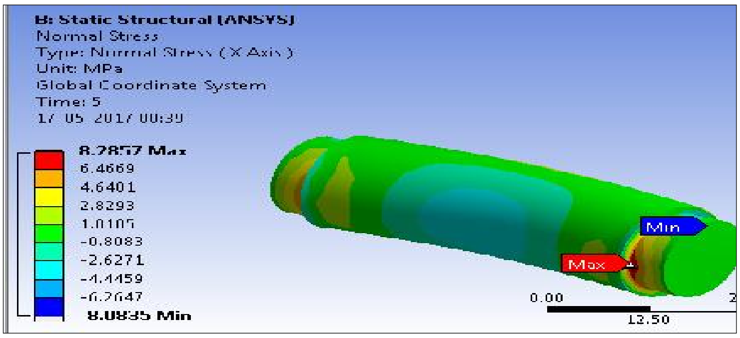

Fig. 9 Normal Stress for Pin

Fig. 9 shows the maximum normal stress which is 8.257 MPa and in safe limit. The with red color shows the area with maximum stress and the region with blue color shows the area with minimum stress.

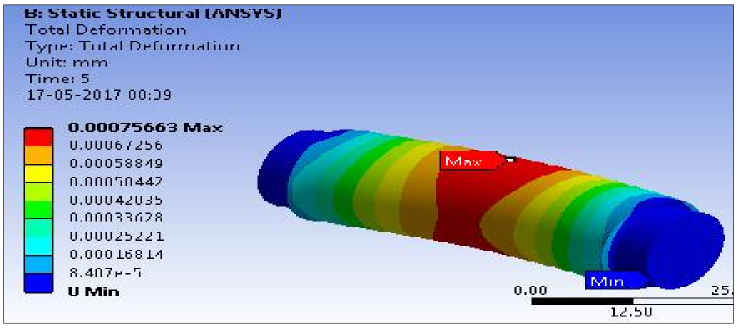

Fig. 10 Negligible deformation pin

Fig. 10 shows the deformation of curved shape link the maximum deformation is 0.00075663 mm which is negligible and in safe region.

EXPERIMENTAL SETUP

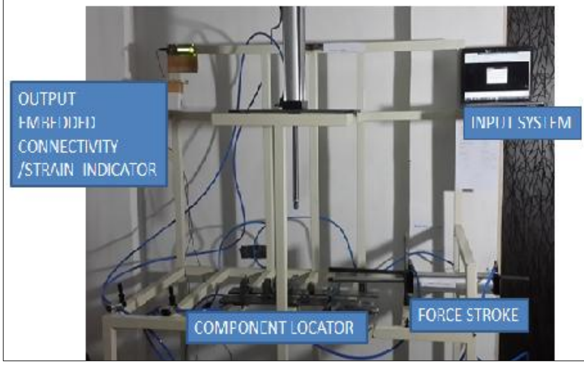

Fig. 11 Experimental Test Setup

Fig. 11 shows the actual view of experimental setup. The name of machine is Semi Automated Strain Gauge Output Analoguer. Strain gauges are fixed on testing specimen by soldering operation. After loading, machine gives strain developed in the specimen. By using stress strain relation, the stress induced in component is calculated. Pneumatic or Hydraulic bar are used for loading.After fixing our specimen into this fixture actual testing is carried out. The blue pipes are seen in this picture are pneumatic pipelines. The strain gauges are fitted on component by soldering operation. Strain gauge mounting is very important operation during experimental validation. If the strain gauges are not mounted properly then reading taken from display will be wrong. For strain gauge mounting surface should be cleaned and well prepared.This is the embedded system in which after loading strain value is displayed on display board. By using pneumatic bar we have to apply force on support element. Input Force = 520 N.

The display shows that the value of strain developed into support element. After getting strain readings by using calculation method find out stress value. Readings are noted from four strain mounts zone after load applied on support element.

Table1. Result Comparison of Analytical and FEA

Parameter	**Analytical**	**FEA**	**Experimental**
Curved Link Chain			
Stress (MPa)	23.31	21	23
Deformation (mm)	0.0061	0.0060	0.0062
Pin			
Stress (MPa)	5.85	5.65	6.1

CONCLUSION

After conducting finite element analysis and experimental validation following conclusions can be drawn.

1. Designed curved chain linkage and pin is found safe according to structural conditions.
2. Maximum total deformation for curved chain and pin is 0.0013 mm and 0.00075 mm respectively which is negligible.
3. Maximum Von-Mises Stress for chain by analytical, FEA and experimental is 23.31, 21, 23 MPa respectively. Maximum Von-Mises Stress for pin by analytical, FEA and experimental is 5.85, 5.65, 6.1 MPa respectively.
4. Stress developed in current chain is less than that of developed in normal chain.
5. By developing transmission drive unit, there is obvious time reduction for operations.
6. This curved chain can be successfully installed as transmission drive element for rotary cage.

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