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FAILURE MODES AND EFFECTS ANALYSIS OF DRIVING MECHANISM IN HORIZONTAL PLANNING MACHINE

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Abstract

In this paper the failure analysis and optimization of horizontal mechanism in a horizontal planning machine rack and pinion arrangement is needed for the translation movement. The relative motion between the fixed and movable bed are being in friction with the ability of V guide ways. And the drive of this mechanism is rack and pinion arrangement. In general, wear occurring in V-Guides ways and drive mechanism are not equal. So that, the variation of the wear amount should affect the surface level of to be machined component. The failure is rectified to implement the new mechanism (arrangement) in driven portion of the horizontal mechanism. The main concept of this paper is that the surface level of the machined component has to be fine.

The modification of the mechanism is to change the position of gear rack and pinion. In existing, the gear tooth is in horizontal and crosswise to the guide ways. Instead of that, the gear tooth is perpendicular to the guide ways. If the wear occurred in this modified one, the gear engaging length will increase. The modified mechanism does not affect the addendum and dedundum of the gear profile.

***Keywords**: Horizontal Planning Machine, Gear Mechanism, Guide Ways.*

1. INTRODUCTION

A planer is a type of metal working machine tool that uses linear relative motion between the work piece and a single-point cutting tool to machine a linear tool path. Its cut is analogous to that of a lathe, except that it is (archetypal) linear instead of helical. A planer is analogous to a shaper, but larger, and with the entire work piece moving on a table beneath the cutter, instead of the cutter riding a ram that moves above a stationary work piece. The table is moved back and forth on the bed beneath the cutting head either by mechanical means, such as a rack and pinion drive or a lead screw, or by a hydraulic cylinder. In transmissions which offer multiple gear ratios, such as bicycles and cars, the term gear, as in first gear, refers to a gear ratio rather than an actual physical gear. The term is used to describe similar devices even when the gear ratio is continuous rather than discrete, or when the device does not actually contain any gears, as in a continuously variable transmission

2. PLANNING MACHINE

The photographic view in Fig.1 typically shows the general configuration of planning machine. Like shaping machines, planning machines are also basically used for producing flat surfaces in different planes. However, the major differences between planning machines from shaping machines are:

Though in principle both shaping and planning machines produce flat surface in the same way by the combined actions of the Generatrix and Directrix but in planning machine, instead of the tool, the work piece reciprocates giving the fast cutting motion and instead of the job, the tool(s) is given the slow feed motion(s).

Compared to shaping machines, planing machines are much larger and more rugged and generally used for large jobs with longer stroke length and heavy cuts. In planing machine, the workpiece is mounted on the reciprocating table and the tool is mounted on the horizontal rail which, again, can move vertically up and down along the vertical rails. Planing machines are more productive (than shaping machines) for longer and faster stroke, heavy cuts (high feed and depth of cut) possible and simultaneous use of a number of tools.

As in shaping machines, in planing machines also,

a) The length and position of stroke can be adjusted

b) Only single point tools are used

c) The quick return persists

d) Form tools are often used for machining grooves of the curved section

e) Both shaping and planning machines can also produce large curved surfaces by using suitable attachments.

Figure: 1 Photographic view of a planning machine

Ebersbach et al[7] In the preceding paragraph, we note that Reference represents the first attempt to consider hybrid manufacturing / remanufacturing systems where machines are subject to random failures and repairs. However, the authors did not address the question of what happens if the machines are used to their maximum production capacity for a long period, and they did not consider the stock of returns. One of the most important results obtained in is the necessary and sufficient conditions for the optimality of the hedging point policy for a single machine, single part-type problem, when the failure rate of the machine is a function of productivity.

It was shown that hedging point policies are only optimal under linear failure rate functions. Their numerical results in the general case suggest that as the inventory level approaches a hedging level, it may be beneficial to decrease productivity in order to realize gains in reliability.

This conjecture has confirmed by the numerical results reported in, where the author considered a long average cost function and a machine characterized by two failure rates: one for low and one for high productivities. Reference generalizes the problem of by considering one machine with different failure rates: more specifically, the failure rate is assumed to depend on productivity, through an increasing function.

Jun-Der Leu, et.al[11] Some of theories or available literature on FMEA methodology is briefly explained as below. In this paper, authors reviewed about different risk evaluation approaches in failure mode & effects analysis. They explained that, the conventional risk priority number (RPN) method along with various risk priority models have been proposed in the literature to enhance the performance of FMEA. In this paper, authors propose a failure analysis method by integrating the Failure Mode Effect and Criticality Analysis (FMECA) and Failure Time Modelling (FTM) based on Proportional Hazard Model (PHM). The objectives of FMECA application are twofold: to classify the censored and uncensored data based on the criticality measure of FMECA, and to identify possible external factors (covariate effects) based on cause and effect assessments of FMECA.FTM based on PHM is applied to analyze statistically the censored and uncensored failure time data by considering the effects of external factors.

3. NECCESSITY OF MACHINE FAULT IDENTIFICATION

Machine fault can be defined as any change in a machinery part or component which makes it unable to perform its function satisfactorily or it can be defined as the termination of the availability of an item to perform its intended function. The familiar stages before the final fault are incipient fault, distress, deterioration, and damage; all of them eventually make the part or component unreliable or unsafe for continued use.

Classification of failure causes are as follows:

1. Inherent weakness in material, design, and manufacturing.
2. Misuse or applying stress in the undesired direction.
3. Gradual deterioration due to wear, tear, stress fatigue, corrosion, and so forth.

Antifriction bearings failure is a major factor in failure of rotating machinery. Antifriction bearing defects may be categorized as localized and distributed. The localized defects include cracks, pits, and spalls caused by fatigue on rolling surfaces. The distributed defect includes surface roughness, waviness, misaligned races, and off-size rolling elements. These defects may result from manufacturing and abrasive wear. Modern manufacturing plants are highly complex. Failure of process equipments and instrumentation increased the operating costs and resulted in loss of production. Undetected or uncorrected malfunctions can induce failures in related equipments and, in extreme cases, can lead to catastrophic accidents. Early fault detection in machines can save millions of dollars on emergency maintenance and production-loss cost. Gearbox and bearings are essential parts of many types of machinery. The early detection of the defects, therefore, is crucial for the prevention of damage and secondary damage to other parts of a machine or even a total failure of the associated large system can be triggered.

There are certain objectives of machine fault identification:

1. Prevention of future failure events.
2. Assurance of safety, reliability, and maintainability of machineries.

Machineries failures reveal a reaction chain of cause and defect. The end of the chain is usually a performance deficiency commonly referred to as the symptom, trouble, or simply the problem. The machine fault signature analysis works backwards to define the elements of the reaction chain and then proceeds to link the most probable failure cause based on a failure analysis with a root cause of an existing or potential problem. Accurate and complete knowledge of the causes responsible for the breakdown of a machine is necessary for the engineer, similarly, as knowledge of a breakdown in health is to the physician. The physician cannot assure a lasting cure unless he knows what lies at the root of the trouble, and the future usefulness of a machine often depends on a correct understanding of the causes of failure. The proper maintenance can be done only after the knowledge of root cause of failure.

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Pravin M. Kinge et al.[1] present a review on fault diagnosis of rotating machinery to provide a broad review of the state of the art in fault diagnosis techniques. The early fault detection and diagnosis allow preventive maintenance and condition-based maintenance to be arranged for the machine during scheduled period of downtime caused by extensive system failures that improves the overall availability, performance and reduces maintenance cost. For the fault diagnosis problem, it is not only to detect faults in system, but also to isolate the faults and find out its causes.

The challenge is to design in quality and reliability early in the development cycle. FMEA is used to identify potential failure modes, determine their effect on the operation of the product, and identify actions to mitigate the failure. A crucial step is anticipating what might go wrong with a product. While anticipating every failure mode is not possible, the development team should formulate as extensive a list of potential failure modes as possible.

3.1 Failure

Figure: 2 Damaged gear

Wear is natural and an inevitable cause that occurs as a result of normal wear aging due to heavy load applied to the planning machine. Wear occurs predominantly in the machine guides ways and hence the rack and pinion tooth suffers damage.

Before modify:

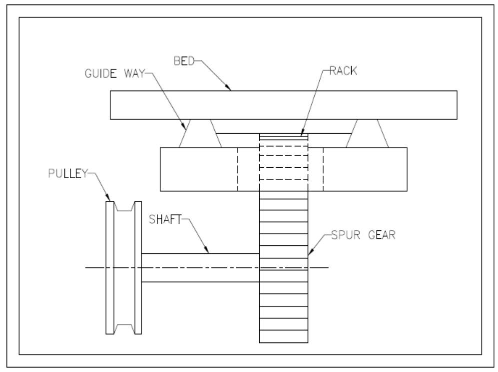

Figure: 3 Line diagram of before modify

After implement:

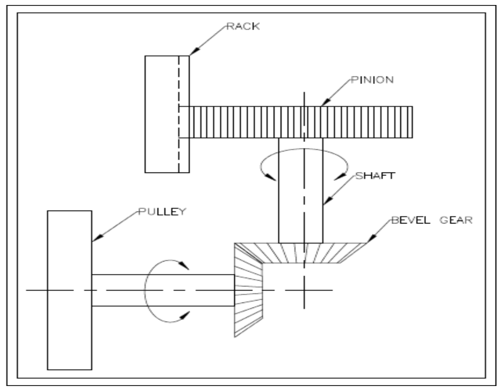

Figure: 4 Line diagram of after implement

3.2 Material Specifications

A type of cast iron with high levels of carbon and excellent compression strength. Gray cast iron is the most common cast iron.

Dr.V.B.Sondur, et.al[2] It is not possible to examine the working parts of a complex machine on load, nor is it convenient to strip down the machine. However, the oil which circulates through the machine carries with it evidence of the condition of parts encountered. Examination of the oil, any particle it has carried with it, allows monitoring of the machine on load or at shutdown. A number of techniques are applied, some very simple, other involving painstaking tests and expensive equipments. Presently, available lubricant sampling or monitoring techniques like rotary particles depositor (RPD), spectrophotometer oil analysis programme (SOAP), Ferrographic oil analysis and recent software used techniques are available to distinguish between damage debris and normal wear debris. Every machine ever designed undergoes a process of wear and tear in operation, yet a battery of modern condition monitoring techniques is available to monitor this process and trigger preventive

maintenance routines which depend on identifying any problem before it has the chance to develop to the point of final breakdown

Density

Gray iron's density makes it excellent for some applications, but inappropriate for others. For example, high density gray cast iron is an excellent choice for pipes, but a cast iron bicycle would be too heavy and too rigid to ride safely.

Density	7.2e-006 kg mm^-3
Coefficient of Thermal Expansion	1.1e-005 C^-1
Specific Heat	4.47e+005mJ kg^-1 C^-1
Thermal Conductivity	5.2e-002 W mm^-1 C^-1
Resistivity	9.6e-005 ohm mm

Table: 1 Gray Cast Iron > Constants

Compressive Ultimate Strength MPa	Compressive Yield Strength MPa
820	80

Table: 2 Gray Cast Iron > Compressive Ultimate Strength

Tensile Yield Strength MPa	Tensile Ultimate Strength MPa
65	240

Table: 3 Gray Cast Iron > Tensile Yield Strength

Temp ºC	Young's Modulus MPa	Poisson's Ratio	Bulk Modulus MPa	Shear Modulus MPa
85	1.1e+005	0.28	83333	42969

Table: 4 Gray Cast Iron > Isotropic Elasticity

Relative Permeability
10000

Table: 5 Gray Cast Iron > Isotropic Relative Permeability

3.3 Design Calculations

Z = 60 teeth

M = 8

Module (m):

$m = D/z$

$8 = D/60$

$D = 8 * 60$

$\therefore D = 480$ mm

Circular pitch (Pc):

$Pc = \Pi D/z$

Where

D = Diameter of the pitch circle

z = Number of teeth on the wheel.

$Pc = \pi \times 480/60$

$Pc = 25.12$mm

∴Pc = 25 mm

Diametral pitch (pd):

Pd = z/D = π/Pc

= π/25

= 0.1256

∴Pd = 0.1256 mm

We are choose the 20° full depth involutes system as per standard proportions of spur gear system. (References from the DESIGN OF TRANSMISSION SYSTEM &PSG data book for page no: 8.22)

Addendum:

Addendum = 1 x m = 1 x 8 = 8mm

Dedendum:

Dedendum =1.25 x m =1.25 x 8 = 10mm

Working depth:

Working depth =2 x m =2 x 8 = 16mm

Minimum total depth:

Minimum total depth = 2.25 x m =2.25 x 8

∴Minimum total depth =18mm

Tooth thickness:

Tooth thickness = 1.5708 x m = 1.5708 x 8

∴ Tooth thickness = 12.5664 mm

Minimum bottom clearance(c):

Minimum bottom clearance = 0.2 x m

= 0.2 x 8 = 1.6mm

Fillet radius at root:

Fillet radius at root = 0.4 x m

= 0.4 x 8.

= 3.2mm

Height factor (f_0):

f_0 = 1

Tip diameter (d_a):

d_a = $(z + 2f_0)$ m

= (60 + 2 * 1) 8 = 62 x 8

∴d_a = 496 mm

Root diameter (d_f):

d_{f1} = $(z - 2f_0)m - 2c$

= (60 - 2 x 1) 8 – 2 x 1.6

= 464 – 1.6

∴d_{f1} = 462.4 mm

Face width:

B = 10 x m

= 10 x8

= 80 mm

W = 2 kW, n_1 = 2950 rpm, Z_1 = 2, Z2 = 32, m = 4mm, $\varphi = 20^o$, d_1= 50 mm, b_1= 65 mm, b_2= 28 mm. Pinion material case carburized steel and Gear material phosphor bronze.

$d_2 = m Z_2 = 4 \times 32 = 128$ mm

Centre distance

$C= 0.5(d_1+ d_2) = 0.5(50+128) =79$mm

Axial pitch: $p = \pi m = 3.14x4 = 12.56$ mm

Lead: $L = p Z_1 = 12.56 \times 2 = 25.12$ mm

Lead angle: $\lambda = \tan^{-1}(L / \pi d_1)$

$= \tan^{-1}(25.12 /\pi \times 50) = 9.09^o$

$V_1 = V_m = (\pi d_1 n_1/60000)$

$= \pi \times 50 \times 2950 / 60000 = 7.72$ m/s

$n_2 = n_1/ i = \{n_1 (Z_2/Z_1)\}$

$= 2950 / (32/2) = 184.38$ rpm

$V_2 = (\pi d_2 n_2/60000)$

$= \pi x128x184.38/60000 =1.24$m/s

$V_S = V_1 /\cos \lambda = 7.72 / \cos 9.09^o = 7.82$ m/s

For V_S = 7.82 m/s and the given materials f = 0.024.

Since the helix angle of the gear is the same as the lead angle of the worm,

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$\Phi_n = \tan^{-1}(\tan\varphi_1 \cos\psi) = \tan^{-1}(\tan 20^o \cos 9.09^o)$

$= 19.77^o$

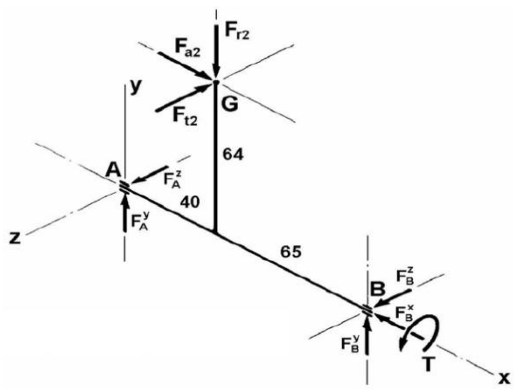
$F_{t1} = W / V_1 = 2000/ 7.72 = 259$ N

$F_{r1} = F^y = F_n \sin\varphi_n = 1503 \sin 19.77^o = 508$ N

$F_{a1} = F^z = F_n (\cos \varphi_n \cos\lambda - f \sin \lambda) = 1503 (\cos 19.77^o \cos 9.09^o - 0.024 \sin 9.09^o) = 1391$ N

Since Bearing B takes the entire thrust load,

FBx = Fa2 = 259 N .



Taking moment about z axis through A, we get

$F_B^y \times 105 – F_{a2} \times 64 – F_{r2} \times 40 = 0$

i.e., $105 F_B^y - 259 \times 64 – 508 \times 40 = 0$

$F_B^y = 351$ N

Σ Fy = 0, from which Fay = 508-351 = 157 N

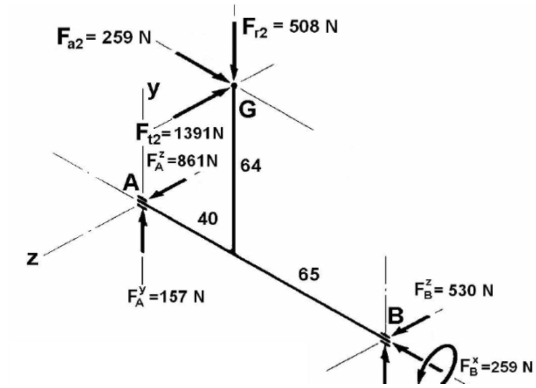
By taking moment about y axis through A, we have

$F_{t2} \times 40 – F_B^z \times 105 = 0$

i.e., 1391 x 40 – 105 FBz = 0 ; $F_B^z = 530$ N

Σ Fz = 0 from which FAz = 1391 – 530 = 861 N

T = Ft2 x r2 = 1391 x 64 x10-3 = 89.02 Nm



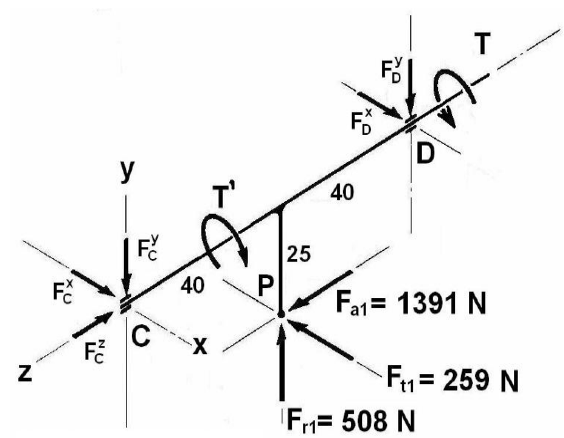
Since the bearing at C takes the entire thrust,

$F_C^z = F_{a1} = 1391$ N.

Taking moment about y (vertical) axis through D,

$F_C^x \times 80 – F_{t1} \times 40 = 0, 80 F_C^x – 259 \times 40 = 0$

$F_c^x = 129.5$ N



$F_D^x = 129.5$ N, since $\Sigma F^x = 0$

Taking moment about x (horizontal) axis through D,

$F_c^y \times 80 – F_{a1} \times 25 - F_{r1} \times 40 = 0$

$80 F_c^y - 1391 \times 25 – 508 \times 40 = 0$

$F_C^y = 689$ N From $\Sigma F^y = 0$, $F_D^y = -181$ N

4. FINITE ELEMENT METHOD

Ansys formulates the equations governing the behavior of each element taking into consideration its connectivity to other elements. These equations relate the displacements to known material properties, restraints, and loads. Next, the program organizes the equations into a large set of simultaneous algebraic equations. The solver finds the displacements in the X, Y, and Z directions at each node. Using the displacements, the program calculates the strains in various directions. Finally, the program uses mathematical expressions to calculate stresses.

4.1 Analysis using ANSYS 14.0

The following figure shows the imported model. In preparing the model for analysis, Ansys subdivides the model into many small tetrahedral pieces called elements that share common points called nodes.

After meshing the model the boundary conditions are applied properly then the final results are obtained.

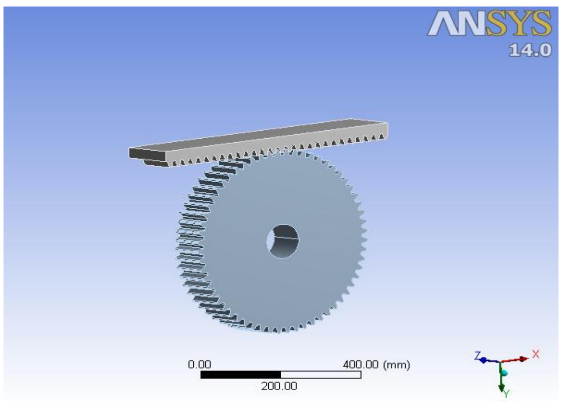


Figure: 5 Design part

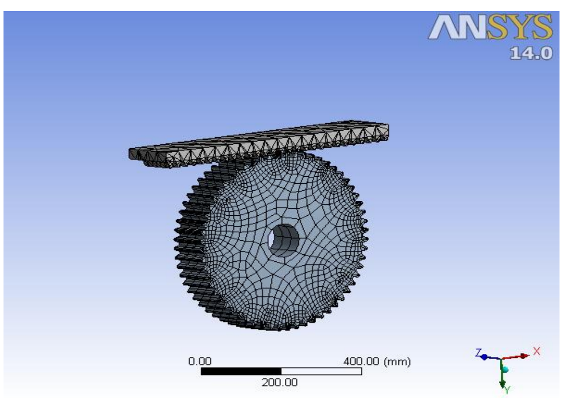


Figure: 6 Meshing part

4.2 Static Structural Analysis

A static structural analysis determines the displacements, stresses, strains, and forces in structures or components caused by loads that do not induce significant inertia and damping effects. Steady loading and response conditions are assumed; that is, the loads and the structure's response are assumed to vary slowly with respect to time. A static structural load can be performed using the ANSYS or Samcef solver. The types of loading that can be applied in a static analysis include:

a. Externally applied forces and pressures
b. Steady-state inertial forces (such as gravity or rotational velocity)
c. Imposed (nonzero) displacements
d.Temperatures (for thermal strain)

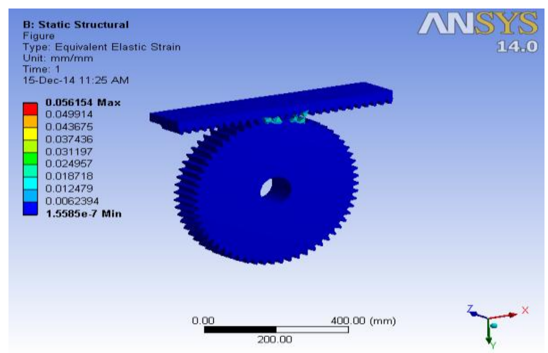


Figure: 7 Equivalent Elastic Strain

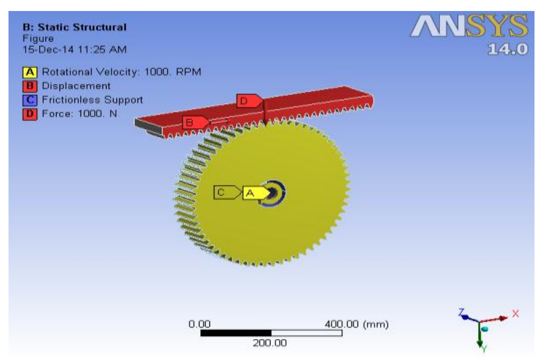


Figure: 8 Static Structural results

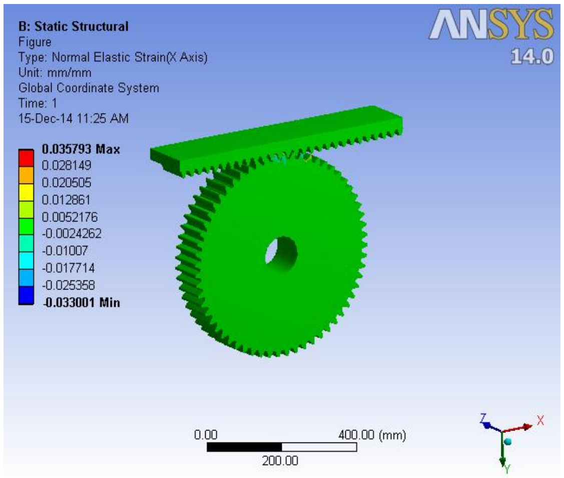


Figure: 9 Shear Stress in XY Plane

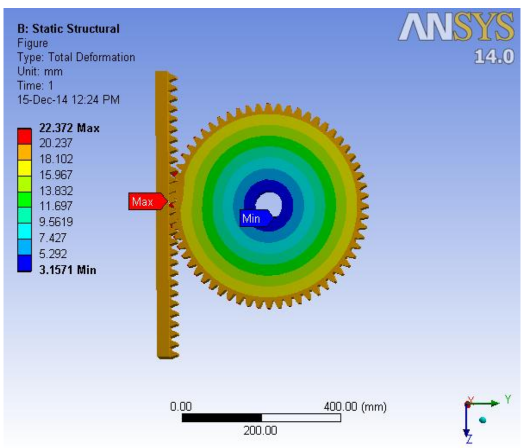


Figure: 10 Total Deformation

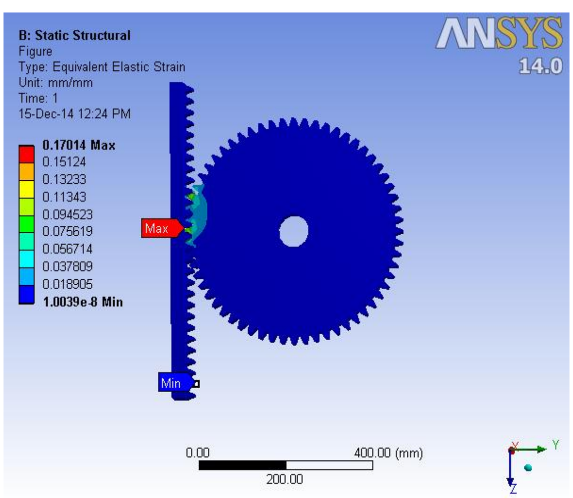


Figure: 11 Equivalent Elastic Strain

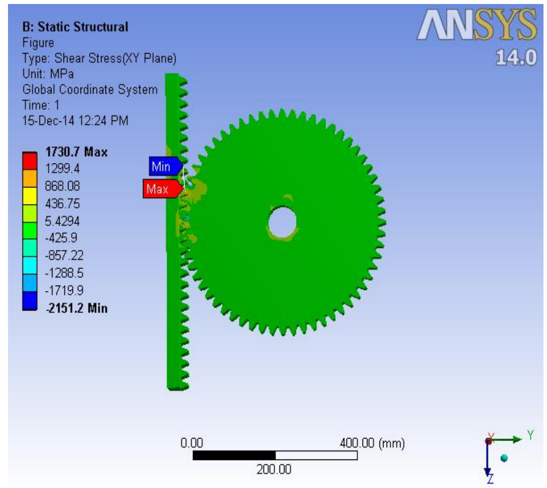


Figure: 12 Shear Stress in XY Plane

5. CONCLUSION

This paper work has provided us an excellent opportunity and experience, to use our limited knowledge. Failure Analysis And Optimitation Of Driving Mechanism of the Horizontal Planning Machine shows satisfactory working environment We are able to understand the difficulties in maintaining the tolerances and also quality. After implementing the new mechanism the Machineries failures were rectified and due to heavy weight work to planning machine the wear occurring in machine V-guides ways were reduced. From the above design calculation and analysis the total deformation and stresses was reduced while comparing the old mechanism

In concluding remarks of this paper, we have developed a **"**Failure Analysis And Optimitation Of Driving Mechanism In Horizantal planning Machine**"**. By using the techniques, they can be modified and developed according to the applications. The problem on horizontal planning machine, gear tooth failure is reduced by implementing the new mechanism (arrangement).

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