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Optimization of Transmission of a Roll Forming Machine using a Planetary Gearbox

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Abstract: - In this thesis, a transmission system of a roll forming machine is designed and optimized using planetary gears. The roll forming machine considered is owned by a company based in Chakan, Pune. A nascent company in the market,"BSS Anuj PVT. LTD." was facing difficulties with a roll forming machine. Out of various the problems, one was of the failing transmission system. Reaching out to the company and gaining their consent, we embarked upon the project of redesigning the transmission system of that machine for the company. The project involves the design of a planetary gearbox and selection of suitable chain drives, replacing the old and conventional transmission system of worm gear boxes. The factors like efficiency and cost of the system were considered while designing. A CAD model of the whole machine was developed using Solid Works and analysis of certain components were analyzed using ANSYS.

Keywords: – *Planetary Gears, Gear Train,Buckingham & Lewis Load, Module, Reduction Ratio.*

I. INTRODUCTION

Roll forming is a metal forming process in which strips of metal are passed through rollers to continuously bend them into the desired cross section. The conventional transmission system of the machine consists of worm gears. These gears are used for the reduction of speed and torque multiplication. But the efficiency of worm gears is low and thus many worm gears have to be used to minimize the losses.

On the other hand, a planetary system higher efficiency. Also it can give very high reduction ratios and the torque density of the gearbox is also very good.

The primary objective of the project is to design the transmission system of a rol forming machine using planetary gears.

II. SELECTION OF GEAR BOX FOR THE OPERATION

The gear design is the most essential part of the entire in the entire design process. The gear design was completed in following stages :

- Deciding Number of Stages and the Gear Ratios for Each Stage.
- Gear Tooth Profile and Gear Type Selection.
- Number of Teeth on Sun Gear.
- Number of Planet Gears in Each Stage.
- Number of teeth on All the Remaining Gears.
- Selection of Material for Each Component.
- Determination of Module for Each Stage.
- Checking for Buckingham & Lewis Load, Wear Load, Bending Stress and Contact Stress.
- Design for Planet and Ring Gears :

A. Deciding Number of Stages and the Gear Ratios for Each Stage:-

The transmission ratio here is 1:40. This is a quite high and so we decide to split the gearbox into two stages. This will ensure that the size of gear will not be very large, making it lighter and easy to move.

We can have the following combination of the gear ratios for each stage –

- 1:10 & 1:4;
- 1:40 & 1:40;
- 1:8 & 1:5.

The ratio is 1:8 and 1:5. Here the torque at the input of first stage is low (121 Nm) , which will negate the size increase due to a large reduction ratio. The second stage will have a high torque input, but since the reduction ratio is small, it will compensate the size increase due to high input torque.

Therefore, we have the following criteria for our planetary gearbox:

Number of stages: 2

- Stage 1 reduction ratio: 8:1
- Stage 2 reduction ratio: 5:1

B. Gear Tooth Profile and Gear Type Selection:-

- We choose full depth involute tooth profile and commercially cut gears. And the pressure angle selected as 20o.
- The type of gear is SPUR GEAR.
- We are using involute profile because it is easy and cheap to cut, pressure angle remains constant throughout the engagement of teeth ensuring a smooth meshing.

C. The No. of Teeth on Sun Gear are:-

$Zs = 2x1 / \sin^2(20)$……. taking f=1 for standard gear tooth and Pressure angle = 20
$Zs = 17.09 = 18$

D. Number of Planets in Each Stage:-

Part	Material	bN/mm2	cN/mm2
Sun gear	40Ni2Cr1Mo28	400	1100
Planet gear	40Ni2Cr1Mo28	400	1100

Table 1

- Lower the reduction ratio, bigger the sun gear and higher the reduction ratio, smaller the sun gear. Consequently, if the sun is small the planets will be larger and when the sun is big, the planets are smaller, for sam ring diameters. It follows that, if there are many big planets around a small sun, the planets may touch each other & there would be an interference among the planets, which is undesirable. Hence, the number planets are selected keeping in mind the sun gear size. Generally, number of planets are less (2-4) when sun gear is small and high (greater than 4) when sun gear is not very small.
- We decide the number of planets here as 3.

E. Number of teeth on All the Remaining Gears:-

For teeth on remaining gears, we have,

CONDITION	CARRIER	SUN	PLANET	RING
Carrier fixed, sun moved by +1 rotation anticlockwise	0	+1	-Zs/Zp	-Zs/Zr
Carrier fixed, sun moved by +x rotation anticlockwise	0	+x	-xZs/Zp	-xZs/Zr
Add +y rotations anticlockwise to all and totalling.	+y	x+y	y- x(Zs/Zp)	y-x(Zs/Zr)

Table 2

We know that when sun makes 8 rotations, the carrier makes 1 rotation.

Thus, we get, $y = 1$; $x+y = 8$
From above equations, we get, $x = 7$.
Since we are keeping ring gear as stationary, $y - x(Zs/Zr) = 0$
Putting values of Zs, x, y, we get, Zr = No. of teeth on ring gear = 126.

From the standard planetary setup with 3 planets, we have the relation,
$Zs + 2Zp = Zr$
Putting Zs =18 and Zr = 126, we get, Zp = no. of eeth on planet gear = 54

Thus, Zs = 18, Zp = 54, Zr = 126

F. Selection of Material:-

We select the following material for the sun gear and the planet:-

For planetary gears, we take = 16
We get, Ys = 0.3245, and Yp = 0.4307 and
[Mt] = motor torque x safety factor = 121 x 1.2 = 145.2 Nm = 145200 Nmm. Since the materials are same, looking at the values of Ys and Yp, the sun is the weaker element.

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G. Module Calculation:-

Minimum module for sun gear = m = 1.8

Since design torque is very high, we take module as 3. Thus, m = 3 mm.

Now, pitch circle diameter of the sun gear is, d = 54 mm and it's width,

b = 16 x 3 = 48 mm. Considering production rate 10 m/min and thus N = 13 rpm,

Linear velocity of the gear is, Vm = 36.75 m/s = 0.612 m/min.

We get, Cv =1.2.

H. Checking for Buckingham & Lewis Load, Wear Load, Bending Stress and Contact Stress.

We now evaluate the following values.

Here, Fs> F (Lewis)...............design safe

Fs>F(Buckingham).....design safe

Fs	F(lewis)	F(Buckingham)	Fw	B	c
18691 N	6453 N	5382 N	12615 N	90.3N/mm^2	1482.7 N/mm^2

Table 4

Fw>F(Buckingham).....design safe

b < [b].....................design safe

c > [c].................design fails for contact stress.

We change the material of sun gear to heat treated Vanadium steel, 50Cr1V23,

[b] =593 N/mm2 and [c] = 1642 N/mm2.

Now the design is safe for contact stresses too.

I. Design for Planet and Ring Gears :-

- *Planets :*

The total tangential force is divided equally on the three planets. We can say that the force at contact points is divided by a factor of 3 when applied. This force numerically equals Ft/3 = 5377/3 = 1792 N. It can be concluded that since the force is lesser on planets, they are safe in design.

- *Ring :*

Ring gear is an internally threaded gear which is stationary and does not transmit power. Also when we calculate for ring gears,in all the equations which have 'i+1' term, that term is replaced by 'i-1'. Due to these two reasons it can be concluded that ring gear is most robust part of system and is safe if made of same materials as the planets.

From above argument we infer that sun is the weakest member in a planetary system and once sun gear is safely designed, we take same material and module to design the rest of the system.

Following are the final dimensions of the first stage in mm.(Module = 3)

	Sun	Planet	Ring
Pitch Circle Diameter	54	162	378

Table 5

Similarly calculations is done for the second stage (Module = 6 mm)

	Sun	Planet	Ring
Pitch Circle Diameter	108	162	432

Table 6

III. DESIGN OF CARRIER

- *Calculation for the first planet Carrier:*

Taking A36 Steel as the Carrier Material.

Syt = 250 N/mm2

$\sigma_c = \frac{250}{3} = 83.33$ N/mm2 (Taking fos = 3)

$\sigma_c = \frac{F}{3 \, x \, 12 \, xt}$

F=8962.96N

t= 2.99 ≈ 3

Minimum Thickness is 3 mm, for better machining purpose we consider it to be as 10 mm.

Diameter of the extended shaft from the Carrier is designed on the basis of shear stress. As it is integrated with the carrier, therefore the material is same.

$\tau = \frac{0.5 \, xSyt}{fos}$

τ =41.67 N/mm2

The output shaft of the carrier is meshed with the second stage sun gear, therefore the shaft is made hollow and design accordingly with the formula given below:

$\tau = 16 \, Mt / \pi \, (D-d)^3$

$(D-d)^3 = \frac{16 \, Mt}{\pi x \, \tau}$

D = 86.04mm

Taking standard value as 90 mm.

- *For the Second Stage Planet gear Carrier:*

The carrier diameter is assumed to be 390 mm as the planet gear's PCD is 135 mm (taking 30 mm pin diameter and 30mm of above clearance).

Taking same Material as that of First stage carrier (A36 Steel)

We get, Thickness of carrier as 30 mm And diameter of extended shaft as 110 mm.

IV. DESIGN OF SHAFT

With reference to the Design of Planetary gearbox, shafts in this gearbox are designed solely on the basis of torque i.e (Maximum Shear Stress Theory)

Torque Transmitted (Mt) = 121 Nm

- *For first Stage:*

Taking Vanadium Steel (50 Cr 1 V 23) as shaft Material

$Syt = 1800\ N/mm^2$

$\tau = 16\ Mt/ \pi\ d^3$

$d^3 = \frac{16\ Mt}{\pi x\ \tau}$

d = 25 mm

- *For second Stage:*

Taking C-45 as material

$Syt = 360\ N/mm^2$

$\tau = \frac{360}{2} = 180\ N/mm^2$

$d^3 = \frac{16\ Mt}{\pi x\ \tau}$

= 136943.986

d = 51.54 mm

Standard Diameter = 55 mm

V. DESIGN OF BEARING

A. *For the First Stage Sun Gear:*

Radius of first stage Sun Gear (rs) = 27 mm.

Torque (T) = 121 Nm.

Speed (n) = 520 rpm.

Diameter = 25mm.

Required: - Bearing number.

Solution:-*Calculations of radial and axial forces*-

Axial force (Fa)= 0 N (since there is negligible axial load in spur gear meshing)

Radial Load (Fr) = Ft Tanα

Here, pressure angle (α) = 20^o

And Tangential load (Ft) = $\frac{T}{r} = \frac{121}{0.027}$ = 4481.48 N

Ft = 4481.48 N

Now, Radial load (Fr) = Ft Tanα

= 4481.48 x tan (20^o)

Radial load (Fr) = 1631.125 N

- *Selection of Type of Bearing Used:*

Since the load is of medium range and mainly there is radial load in action, Deep Groove Ball Bearing can be used.

Deep Groove Ball Bearing has four different series named as Series 60, Series 62, Series 63 and Series 64 with 90% reliability.

- *Determine values of Radial and thrust factors, Service Factor (S):*

From PSG Design Data Book, X= 1, S=1.3, Kt = 1

- *Calculate the equivalent load from the equation:*

P = (X.V. Fr + Y.Fa). S.Kt

Here, X = 1, V = 1, Fr = 1631.125N, Fa= 0 N, S= 1.3, Kt= 1

P = (1x1x1631.125 + 0) x1x1.3

=2120.46 N

Assuming life of bearing to be 30,000 hours.

Life in million rotations (L_{10}) =60 x L_{10h} x n/ 10^6

$= \frac{60\ X\ 30000\ X\ 520}{10\hat{}6}$

L_{10} = 936 mr

- *Calculate dynamic Capacity from the equation:*

C= P $(L_{10})^{1/3}$

Here,

P = 2120.46 N

L_{10}= 936 mr

C = 2120.46 x $(936)^{1/3}$

= 20742.22 N

C = 2074.22 kgf

Checking with the Load carrying Capacity of the Standard Bearing from the Design Data Book:

Bearing no. = 6405

d = 25mm

Dynamic Capacity = 2600 kgf.

Deep Groove Ball Bearing no. 6405 is selected.

B. *For the First Stage Carrier :-*

- *Data:-*

Torque (T) = 968 Nm.

Speed (n) = 65 rpm.

Diameter = 90 mm.

Radial load (F_r) = 7829.40 N

P =10178.22 N

L_{10} = 117 mr

C = 4978.14 kgf

Bearing no. = 6218

d = 90mm

Dynamic Capacity =7500 kgf.

Deep Groove Ball Bearing no. 6218 is selected.

C. *For the second stage carrier:-*

- *Data:-.*

Torque (T) = 968 Nm.

Speed (n) = 65 rpm.

Diameter = 110mm

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Radial load (F_r) = 6405.87 N
P = 8327.631 N
C = 1832.244 kgf

Bearing no. = 6022
d = 110 mm
Dynamic Capacity =6400 kgf.

Deep Groove Ball Bearing no. 6022 is selected because of the required bore diameter.

D. For the Output Shaft :-

- *Data:-.*
  Torque (T) = 4840 Nm
  Speed (n) = 13 rpm
  Diameter = 55 mm

As the output Shaft has the same dynamic load capacity as that of the second stage carrier so by referring Design Data Book Bearing no. 6011 is selected.
Diameter of bearing (d) = 55 mm
Dynamic Load Capacity (C) = 2200 kgf

VI. DESIGN OF CHAIN

Considering the maximum production rate of 10 m/min, chain speed is 13 rpm.

Corresponding input power = torque x rpm = 6.5 kW

Selecting the correction factor as 1.3, we get,

Design Power = [P] = 1.3 x 6.5 = 8.45 kW.

Here, transmission ratio = 1, and so both sprockets are identical.
From the kW rating chart, we get the chain as RS-180 and number of teeth on sprocket as 14. So, Z = 14.

With reference to TSUBAKI ROLLER CHAINS company's American standard chain catalogues,
Pitch of the chain = p = 57.15 mm
Centre distance between two sprockets = 515.6 mm

Diameter of the sprocket = psin180Z= 256.83 mm

The sprocket diameter is within the size restrictions and will not obstruct or touch stand or any other part of the machine.

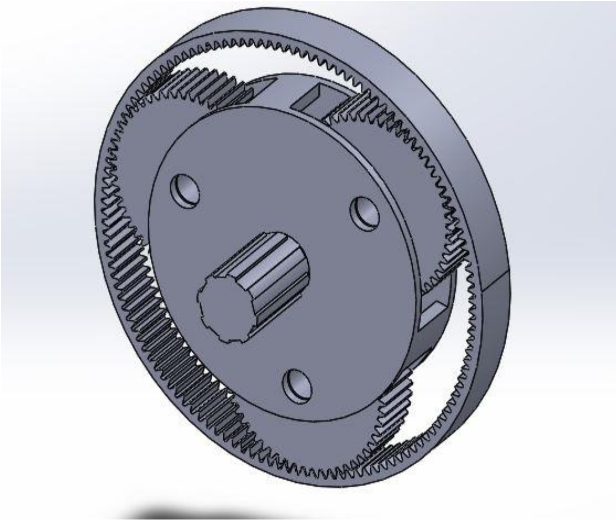

Fig 1:- Stage 1 Assembly

Fig 2:- Stage 2 Assembly

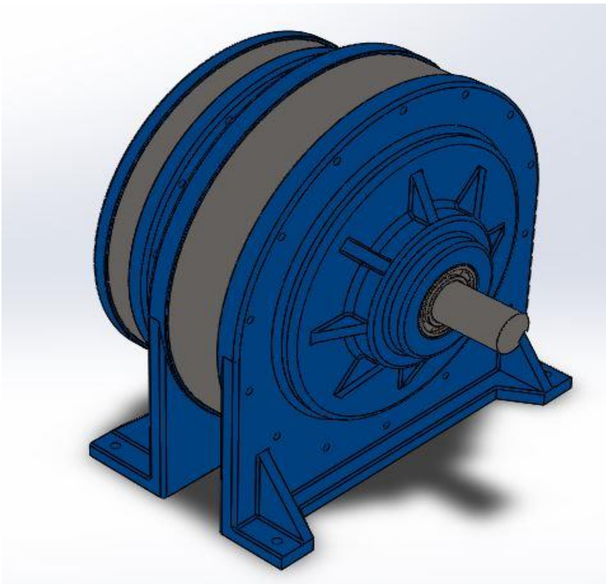

Fig 3:- Planetary Gearbox

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VII. RESULT OF ANALYSIS

➢ *Stage 1 Analysis:-*

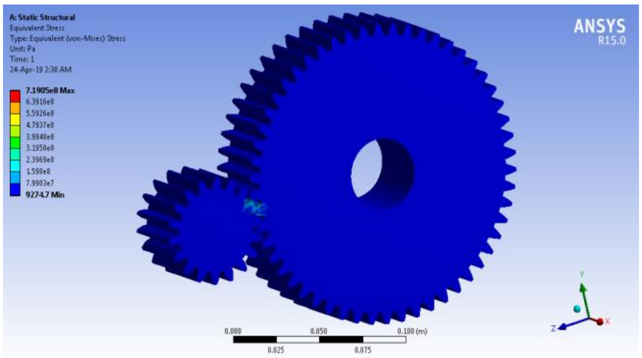

Fig 4:- Contact Stress Results

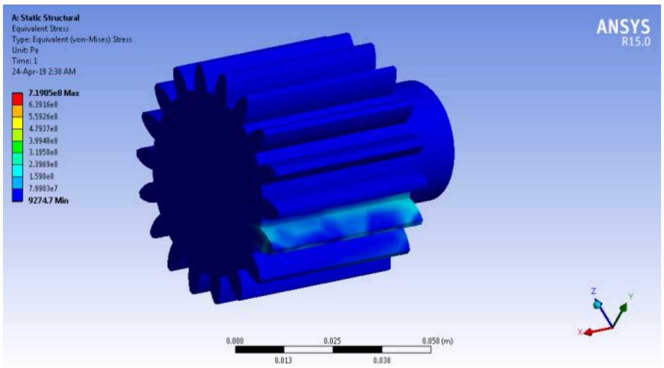

Fig 5:- Stress Distribution on Sun Tooth

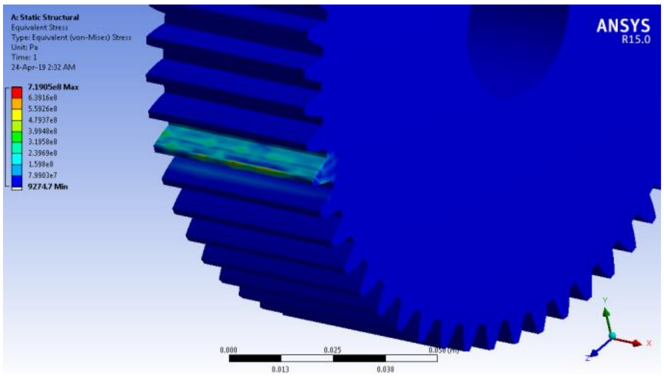

Fig 6:- Stress Distribution on Planet Gear Tooth

➢ *Stage 2 Analysis:-*

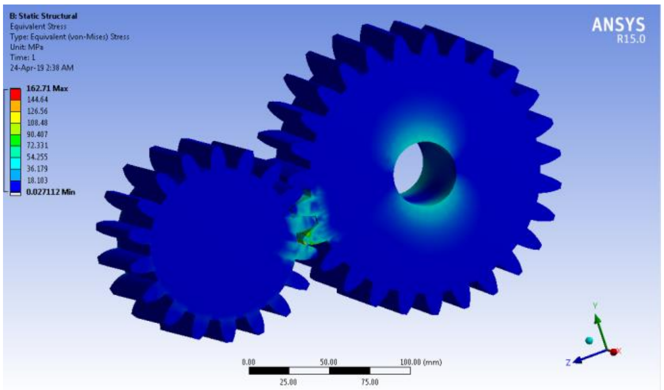

Fig 7:- Contact Stress Results

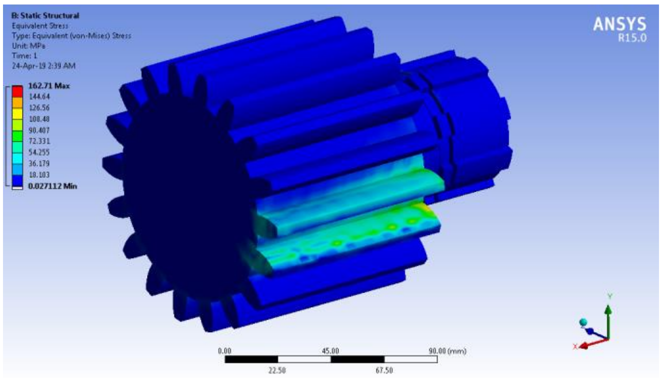

Fig 8:- Stress Distribution on Sun Gear Tooth

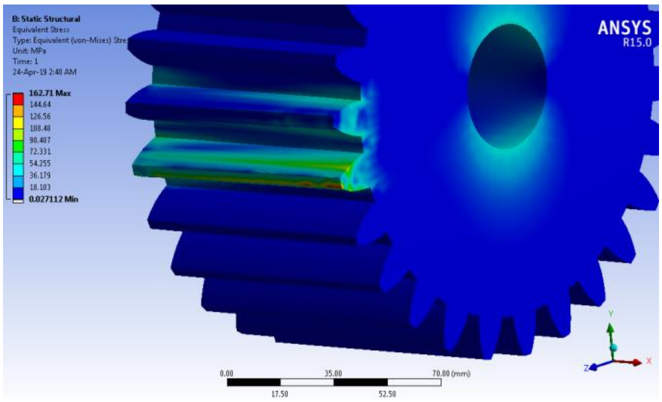

Fig 9:- Stress Distribution on Planet Gear Tooth

➢ *Stage 1 Carrier :-*

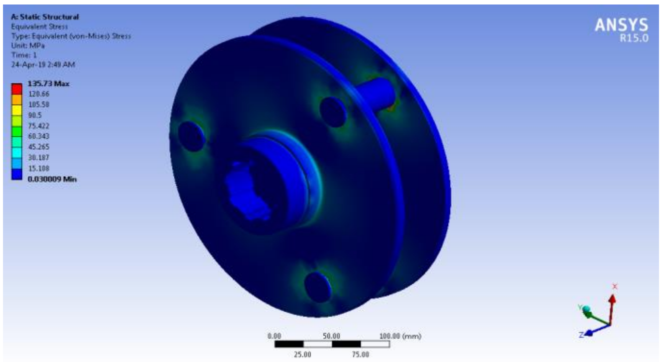

Fig 10:- Stress Distribution

➢ *Stage 2 Carrier:-*

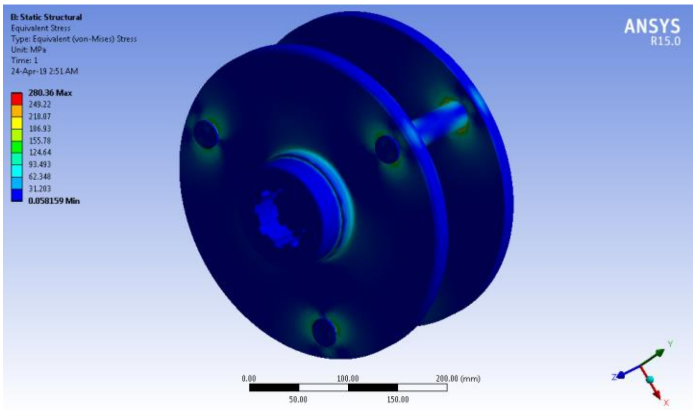

Fig 11:- Stress Distribution

➢ *Casing:-*

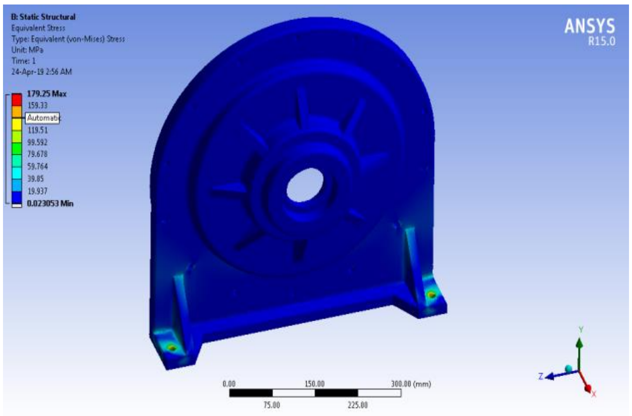

Fig 12:- Stress Distribution

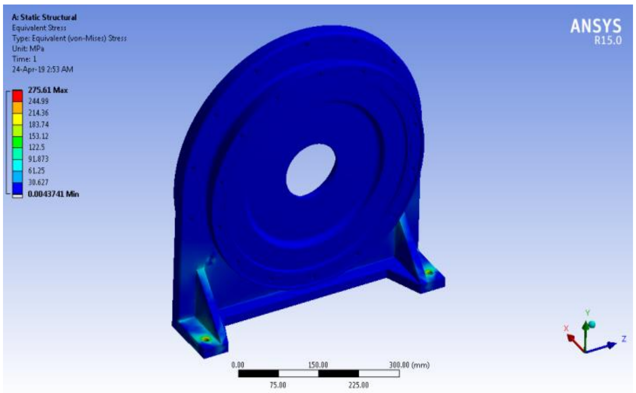

Fig 13:- Stress Distribution

VIII. CONCLUSION

Following are the major conclusions from the project

- A new and workable form of transmission system system has been designed for roll forming machines.
- It has been proven that conventional worm and worm wheel drive is not the only transmission system possible for roll forming machine.
- The planetary gearbox has been designed and analysed keeping in mind, the practical conditions.
- The chain, bearings, shafts and the casing have been designed and are safe.
- The new transmission system, containing planetary gears iare more efficient.
- The cost of the transmission system has been reduced considerably. The multiple worm gears which cost a vast sum of money have been replaced by a single gearbox. Since efficiency has increased, there is less power consumption than the worm gear transmission system.
- The faulty chain drive has been redesigned and is safe.

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