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COMPUTER AIDED DESIGN AND ANALYSIS OF ROTOR SHAFT OF ROTAVATOR

Mr. S. A. Mishra
M-Tech. scholer
Dept. of Mechanical Engineering
B.D.College Of Engineering, Sevagram, Wardha, India
Saurabh.mishra285@gmail.com

Dr . A. R. Sahu
Professor
Dept. of Mechanical Engineering
B.D.College Of Engineering Sevagram, Wardha, India
Anilrsahu50@gmail.com

Prof. R. D. Thakre
Asso. Professor
Dept. of Mechanical Engineering
B.D.College Of Engineering Sevagram, Wardha, India
rdthakre@rediffmail.com

Prof. U. D. Gulhane
Professor
Dept. of Mechanical Engineering
B.D.College Of Engineering Sevagram, Wardha, India
udgulhane@rediffmail.com

ABSTRACT –

Rotavator is important agricultural equipment playing significant role in seedbed preparation. Now a days most of farmers preferring rotavator for the seedbed preparation than traditional methods. There are different types of rotavators are to be used for different soil conditions. Now due to improper knowledge of scientific design and material selection the rotavator get fail after some time. Different manufacturers removed the problem of failure of different parts of rotavator without implementing any design process. In this paper by carrying out extensive literature review, different reasons of failure of rotavator have been identified. The existing rotavator is analyzed by doing modeling and carrying out ANSYS analysis. It has been found out that the material for flange is not proper to withstand the different types of forces incurred on it. Therefore new material EN 19(oil Quenched & drawn) has suggested.

Keywords—Rotavator, rotor shaft, Flange, Blades

I. INTRODUCTION

Automation in industry gaining more importance and popularity, since it helps to reduce cost as well as time with increased productivity and quality, the automation is becoming more popular in agricultural field as well. This resulted into world class facilities for agricultural crop from sowing to harvesting to storage. Farmers usually take two or more crops in a year. In sequential cropping system, seedbed is required to be prepaired for the next crop. Land development is costlier affairs in farming and fro this agricultural equipment i.e. Rotavator is used now a days. A Rotavator is agricultural equipment which is capable of doing job of tillter, disc horrowand leveler. Rotavator destroys the weeds, stubbles of paddy, sugarcane and mize crops completely. It saved 30-35% of time and 20-25% in the cost of operation as compared to tillage by cultivator.

MAIN PARTS OF ROTAVATOR

1. Independent Top Mast: one end of shaft will be connected to tractor P.T.O. and another end to rotavator.

2. Single / Multi Speed Gear Box: A gear box with bevel gears, main shaft, pinion shaft, heavy duty roller bearings combine form a unit to reduce standard P.T.O. rpm 540 rpm to 204 rpm. It enables the rotor shaft to rotate in the direction of travel.
3. Chain / Gear Cover Part Flange: A chain and gear cover part flange is a supporting element on which chain and gears are mounted.
4. Blades: The L shaped will be most common due to L shape is usually superior to others in heavy trash.
5. Chain / Gear Cover Part: A chain and gear cover part is a covering element in which chain and gears are safely protected from outside.
6. Frame and Cover: By adjusting the position of rear cover; the degree of pulverization of soil will be controlled.
7. Adjustable depth skids: It is fixed on adjustable frame to fix up a distance a gap between soil and Blade contact i.e. depth skid.
8. Offset adjustable frame: There is fixed rigid support to side parts mounted on rotary blade mounted shaft.

Fig.1 Main Parts of Rotavator

II. PROBLEM IDENTIFICATION

At the time of project selection it is observed that there are frequent failures of parts of Rotavator, such as Blade, Nut & Bolt assembly & Flange. But it is important to note that the other parts of rotavator easily available and it is easy and convenient to repair or to replace them, but if there is any failure in the Flange of the rotavator than the owner has to replace the entire Rotor Shaft Assembly which is much costlier & time consuming to both Rotavator owner and manufacturer. It is observed that in normal conditions there is no failure in Rotavator, but when speed of tractor is increases or some obstacles such as Roots of Plants, Hard stone, Weed etc. comes in the path of rotavator while working, more torque is required. To overcome those obstacles and due to that there is chance of failure of rotavator shaft.

Fig.2 Failure of Rotor Shaft (Flange)

III. Design Calculations For Rotor Shaft Of Rotavator

The power from tractor is transmitted to Rotavator through PTO driveshaft. Thus for calculating the forces and stress acting on the Rotor shaft and Blade, specification of Tractor and rotavator must be known. Project point of view a tractor rotavator system is considered for the study, specification of which is as given below.

Table 1: Specification of Tractor Rotavator System.

Sr.No.	Parameter	Description
1.	Size of Rotavator	1.75m (Medium Size)
2.	No. of blades on Rotavator	42
3.	Rotational speed of Rotavator	246 RPM
4.	Overall weight of Rotavator	475 Kg.
5.	Tractor Required for Processing for Medium size Rotavator	45HP to 50HP.
6.	Overall length of Tractor	133.9”
7.	Width of Tractor	1.45M
8.	Height of Tractor	2.38m

Table2: Material Specification for Rotavator & Its Main Components

Sr. No.	Components	Material
1.	Rotavator Shaft	EN19
2.	Flange	SAE 1020 (AISI 1020)
3.	Blades	Boron Steel Hardened & Tempered
4.	Bolts	8.8 grade (AISI 5140)
5.	Gears	EN353

Assumptions

1. The design and modification is carried out by considering the above specification of the rotavator.
2. Tractor Capacity 45HP @ 2300 RPM.

(1) Time Required for Processing of Land

A Medium size Rotavator Process 0.6 Hector area in 1 hour i.e. 60 min.

Therefore total time for processing = **3600 Secs.**

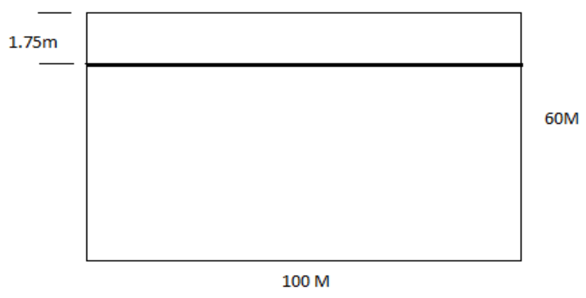

Fig .3 Land Area Preparation

Velocity Calculations

No. of rounds required for a medium size Rotavator = 60 / 1.75

= **34.28 ≈ 35 nos**.

Now, distance travelled by the Rotavator in one run = 100 m

Therefore total distance travelled by the Rotavator = 35 × 100

= **3500 m**

Velocity of tractor = Distance Travelled / Time Taken

= 3500 / 3600

= **0.972 m/s.**

= **58.32 m/min.**

Power Required for Workdone

For the use of a standard medium size Rotavator a 45 HP tractor is required.

Therefore engine power of Tractor = **45 HP.**

Considering 20% Transmission losses, therefore power available at PTO shaft = **36 HP.**

Therefore,

Power required for the implementation = Power at PTO shaft – Power required for propulsion of tractor

Now,

Power required for the propulsion of tractor (HP)

$$= \frac{(16.9\,V^2 + 0.032\,V^2)\,W + 0.03212\,AV^2}{269.95}$$

Where,

A – Projected area of tractor (B × H).

V – Velocity of tractor in m/Min.

W- Weight of tractor with complete rotor assembly.

Area of Tractor A =1.45 × 2.38

= **3.415 m^2**.

Weight = weight of Tractor + Weight of PTO drive Shaft + Weight of Rotavator

= 2010 + 5 + 475

= **2490 Kg =2.49 Tonn.**

Therefore HP

$$= \frac{(16.9 \times 58.38^2 + 0.0329 \times 58.38^2)\ 2.49 + 0.03212 \times 3.451 \times (58.32^2)}{269.95}$$

= **11.49 HP**.

Therefore, power required to drive the Rotavator =

Power at PTO shaft – power for the propulsion of tractor.

= 36 -11.49

= **24.51 HP**.

= **18.28 KW**.

Torque & Force Calculation

Now, the Torque generated at the Rotor shaft (T) $= \frac{60\ P}{2\ \pi\ N}$

$$= \frac{60 \times 18.28 \times 10^3}{2\ \pi \times 246}$$

T= **709.59 N-m**

= **709.59 × 10^3 N-mm.**

Therefore, Force acting on blade due to torque (F) = **Torque / R**

R- Perpendicular Distance of blade with soil. = 0.274 m

F = 709 59 / 0.274

= **2589.77 N**

Therefore force acting per flange = 2589.77 / 8

= **323.72 N**.

Angular velocity of rotor shaft = **ω** = $\frac{2\pi N}{60}$

$$= \frac{2\ \pi \times 246}{60}$$

= **25.76 Rad /Sec.**

Now the Torque on Rotor Shaft at the starting = I × α

Where I- Mass Moment Inertia = 3.599 Kg-m2.

α- change in angular Acceleration = $\frac{\omega 2 - \omega 1}{dt}$

Where,

dt- Time To Reach Final Velocity = 3.5 sec. for 0- 246 RPM

N1- 0 RPM. N2- 246 RPM

And, dt = 2 sec. for 246 – 210 RPM.

ω_1 - angular velocity at 0 RPM & 246 RPM respectively.

Therefore $\alpha 1 = \frac{25.76}{3.5}$

$= \mathbf{7.36\ Rad/sec^2}$

Therefore torque at starting **= I × α1**

= 3.599 × 7.36

= **26.51 N-m.**

Now, calculating Torque & Force acting on blade at the time of processing.

At the time of processing the working speed of Rotor shaft is 195 to 210 RPM.

The taking Avg. speed = 195 + 210 /2

= 202.5 RPM.

Here

N1= 246 RPM therefore ω_1 = 25.76 Rad.

N2 = 202.5 RPM ω_2 = 21.20 Rad.

D_t = 2sec.

Therefore, $\alpha 2 = \frac{\mathbf{25.76-21.20}}{\mathbf{2}}$

$= \mathbf{2.28\ Rad/sec^2}$

Torque available at time of working = I × α2

= 3.599 × 2.28

= **8.205 N-m**.

Now, **Soil resistance pressure = 0.56 Kg/cm2.**

The Force exerted by soil on Blade = Pressure × Blade surface Area

Area = (14 × 8) +1/2 (8 × 8)

Area = **144 cm^2.**

Therefore the resistive Force **(F_R)** = 144 × 0.56

= **80.64 Kg**.

= **80.64 × 9.81**

F_R= 791.07 N-m.

Therefore, the Torque required at the time of processing = Resistive Force by Soil × (Blade Length + Disc Length)

=791.07 × (0.274)

=**214.38 N-m.**

Stiffness of Blade material K = **5.26 ×10^6 N/m**

Considering the Impact loading on the Rotor Shaft at the various Angular Speed.

The weight of rotor shaft Assembly = 128.4 Kg

∴ The mass of Flang & Blade Assembly = 128.4 / 8

= 16.05 Kg.

For the Impact Loading

$$F = \sqrt{(I\omega^2 K)}$$

Where, F –Impact Force Due to Angular speed (N).

I – Mass moment Inertia of Individual Flange ($Kg.m^2$).

ω –Angular Velocity of Rotor shaft (rad/Sec).

A standard Tractor propelled rotavator, The Linear speed of tractor is in forward direction. And the Rotor shaft speed is rotating at speed range of 190 RPM to 210 RPM.

Therefore to calculate the Impact Forces at different angular speed, so we consider the Rotational speed at three stages i.e. 190 rpm, 200 rpm, 210rpm.

$$\therefore \omega_1 = \frac{2\pi N}{60} = \frac{2\pi \times 190}{60}$$

= **19.89 rad/sec.**

$$F = \sqrt{0.44 \times 19.89^2 \times 5.263 \times 10^6}$$

F = 30267.58 N.

∴Total Force Ft = F + F_R

= 30267.58 + 791.07

= 31058.65 N

Similarly calculating the total forces At various different angular speed.

Table No.3 Force And Speed Table

Speed of Rotor Shaft(RPM)	**Angular Velocity, ω (rad/Sec.)**	**Impact Force F (N).**	**Total Force Act bon Blade (F+F_R), N**
190	19.89	30267.58	31058.65
200	20.94	31865.42	32656.46
210	21.99	33463.26	34254.33

Due to the sudden Impact loading, the Flange on Rotor shaft will get bend.

∴Checking permissible Bending stress for flange material SAE 1020 (AISI 1020)

S_{yt} = 246 N/mm2 S_{ys} = 154 N/mm2

Assuming Factor of safety (**Fs**) = 1.5

∴ Permissible Bending Stress σ_b = S_{yt} / Factor of Safety

= 246/1.5

σ_b = 164 N/mm2

We know that

$\sigma b = \frac{M}{Z}$

Where,M–Maximum Bending moment= Force×Perpendicular Distance

Z – Section modulus of Flange. = $\frac{bh2}{6}$

b = 110mm h = 7 mm

σb – Induced Bending Stress.

$Z = \frac{110 \times 7^2}{2}$

Z = **898.33 mm4**

M = Force × Perpendicular distance

=31058.65 × 170

= 5145.47 × 10^3 N-mm.

Since the Bending Stress are acting in longitudinal direction, therefore taking 20% value for bending forces

$\therefore M = 5145.47 \times 20\%$

M =1029.09 × 10^3 N-mm

Therefore Induced Bending Stress = $\frac{1029.09 \times 10^3}{898.33}$

σ_b= 1045.56 N/mm2

Similarly calculating for all values

Table No.4 Force, Moment, Section Modulus & Induced Stress Table

Speed of Rotor Shaft (RPM)	Impact Force F (N).	Total Force Act on Blade (F+ F_R), N	Total Moment (M), N-mm	Longitudinal Moment (20%M)	Section modulus Z, mm4	Induced σ_b N/mm2
19.89	30267.58	31058.65	5145.47×10^3	1029.09 ×10^3	898.33	1145.56
20.94	31865.42	32656.46	5557.60×10^3	1110.32 ×10^3	898.33	1235.98
21.94	33463.26	34254.33	5823.23×10^3	1164.64 ×10^3	898.33	1296.45

IV. Modeling Of Rotor Shaft Of Rotavator

Modeling is a process of generating three dimensional objects of the real world for the purpose of designing, analyzing, drafting and manufacturing. Modeling creates a data base in the computer which represents the object generated. This object database is used to display the object, to prepare drawings of the object with different views, to prepare data for analysis and design.

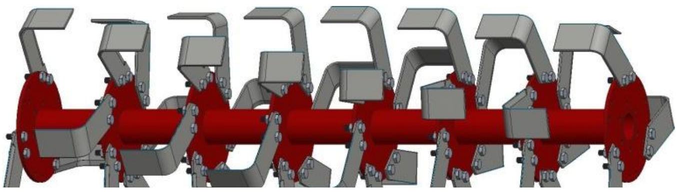

Fig.4 Modeling of Rotorshaft of Rotavator

V. ANALYSIS OF EXISTING AND MODIFIED ROTOR SHAFT BY ANSYS 14.0

The material given for the Shaft, Flange, Blades and Bolt is EN 19, SAE 1020(AISI 1020), Boron Steel, Grade8.8 (AISI 5140) respectively. The properties of these materials are given in ANSYS and Creo model is meshed, & then Performed the Finite ELEMENT Analysis on the Flange Assembly.

After creating a mesh, applying a force on the Blade of Flange assembly. Here we observed that, at the time of working of a rotavator that the Blades mounted on the Flange are achieving certain depth in the soil while processing, thus at the time when shearing action take place the impact force will exert be exerted on the blade in all the direction. Thus here for analysis purpose we assumed that this forces are acting in a certain ration i.e. 70% in X-direction, 20% in Y-Direction, and 10% in Z- Direction, due to which the flange mounted on the blade will get bend. After conducting the analysis following results were observed

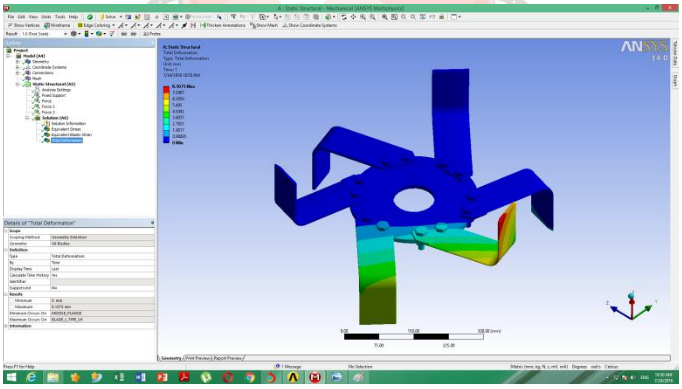

Fig. 5 Total Deformation of blade (6mm) & Flange (3.21mm)

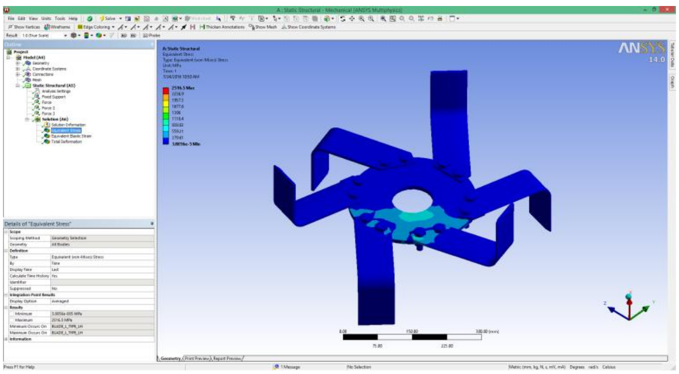

Fig. 6 Total stress acting on the Assembly (838 MPa)

By performing the analysis of Existing flange of rotor shaft of rotavator it is found that, at angular velocity of 19.89 rad/sec, it will fail. Because the stress induced in the flange material are more than the permissible stress values of material AISI 1020. It is also found that the deflection of flange is 3.21mm. So we decide to change the material of flange to EN 19 (Oil Quenched & Hardened).

VI. Analysis of modified rotor shaft

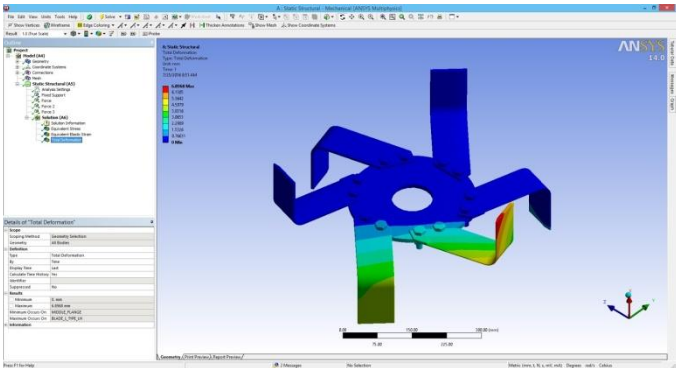

Fig. 7 Total Deformation of blade (6.8mm) & Flange (1.52mm)

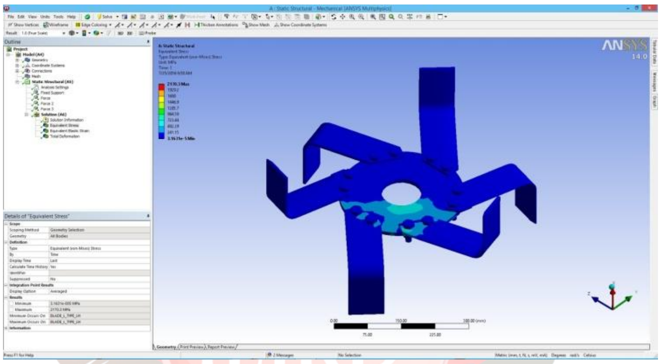

Fig. 8 Total stress acting on the Assembly (482.29MPa)

VII. Fatigue life analysis

The three Major Life methods used in design and analysis are the stress-life method, the strain life Method and the linear elastic fracture mechanics method. These methods attempt to predict the life in number of cycles to failure, N, for a specific level of loading. Life of **$1 \leq N \leq 10^3$** cycles is generally classified as **low – cycle fatigue**, whereas **high-cycle fatigue** is considered to be **$N > 10^3$Cycles.**

The stress Life Method

To determine the strength of materials under the action of fatigue loads, specimens are subjected to repeated or varying forces of specified magnitudes.To establish the fatigue strength of a material, quite a number of tests are necessary. Because of statistical nature of fatigue. The first test is made at a stress that is somewhat under the ultimate strength of the material.

The second test is made at a stress that is less than that used in the first. This process is continued, and the results are plotted on S-N-Diagram. This chart may be plot on semi-log paper or on log-log paper.

The ordinate of the S-N-diagram is called the fatigue strength Sf;a statement of this strength value must always be accompanied by a statement of the number of cycle N to which it correspond

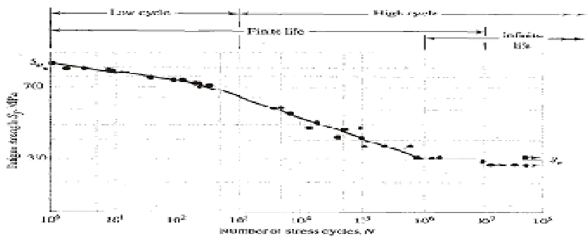

Fig.6.12 S N Curve

Steps for Fatigue Calculation

$Se = K_a . K_b . K_c . K_d . S'_e$

Where K_a – Surface finish Factor = $a.(Sut)^b$ a= 4.51; b= -0.265

K_b – Size Factor =0.75

K_c – Relibility factor = 0.75

K_d – Modifying Factor =$1/K_f$ = 1

S'e = Endurance limit stress of a specimen subjected to reversible bending. = $0.5(S_{ut})$

Fatigue Stress Calculation for AISI 1020

- Sut = 435 Mpa (From Design Data Book)
- S'e= 0.5(435) =217.5 Mpa
- a= 4.51
- b= -0.265
- K_a = 0.9051
- Se=131.91 MPa

Fatigue Stress Calculation for EN 19 (Oil Quenched & Drawn 425^{0}C)

- S_{ut} = 1300 Mpa (From Design Data Book)
- S'e= 0.5(1300) = 650 Mpa
- a= 4.51
- b= -0.265
- K_a = 0.763
- Se= 294.73 MPa

Fatigue Life Cycle Calculation For Material EN19

To calculate the life cycle of Given Material First we have to plot the respective stress values on S-N curve.

Log_{10} (S'e) = Log_{10} (1170)
= 3.068
Se =294.73 N/mm^2

Log_{10} (Se) = Log_{10} (294.73)

= 2.46

Sf = Log_{10} (σ_b) = Log_{10} (723.33)

Sf = 2.85

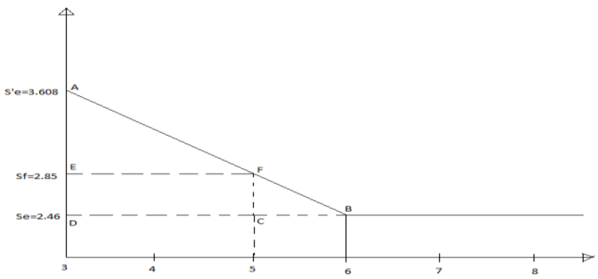

Fig.9 Actual Plotting of S-N Curve For EN19

Now finding the number of cycles for material

$$EF = \frac{DB + AE}{AD} = \frac{(6-3)(3.608-2.85)}{(3.608-2.46)}$$

EF = 1.9808

Log_{10} N = 3 + EF

Log_{10} N = 3 + 1.9808

N = 95675.33 Cycles.

Fatigue Life cycle Analysis

In Fatigue analysis We have to Consider the Common Cycles, For Material of AISI 1020 & EN 19. Then From the Two analyses we compare the wear rate of AISI 1020 and EN 19.

VIII. Result and Discussion

Analysis of existing Rotor shaft is done at maximum loading conditions, and Following results are obtained.

Table no.6 Total Deformations and Maximum Shear Stress For Material AISI 1020

Sr. No.	Angular Velocity (Rad/ Sec)	Total Forces(Impact + Soil Resistivity) N	Deformation Of Flange (Mm)	Maximum Shear Stress (Mpa)
1	19.89	31058.65	2.41	733.33
2	20.94	32656.52	2.44	788.31
3	21.99	34254.33	2.70	838.82

From the above analysis it is found that the maximum bending stress on the flange is more than the allowable stress of the material AISI 1020 at a angular velocity of 19.89 rad/sec. The force acting at this speed is 31508.65 N and the deformation of flange is 2.41mm and maximum bending stress acting on flange is 733.33 N/mm^2 and it is increasing as the forces are increasing on the blade. So we change the material of flange to En 19 which has more allowable stress than the AISI 1020.

Table no.5 Total Deformations and Maximum Shear Stress for Material En 19(Oil Quenched & Drawn 425^{0}C)

Sr. No.	Angular Velocity (Rad/ Sec)	Total Forces(Impact + Soil Resistivity) N	Deformation Of Flange (Mm)	Maximum Shear Stress (Mpa)
1	19.89	31058.65	1.53	428.89
2	20.94	32656.52	1.64	525.53
3	21.99	34254.33	1.78	551.20

After changing the material of flange it is observed that when the angular velocity of 21.99 rad/Sec and force 34254.33 N, the deflection of flange is 1.78 mm and the maximum bending stress acting on flange is 551.20 N/mm^2. So it is suggested to use as EN 19 as the flange material to avoid the failure of flange and blade assemble. So from the above analysis it is observed that the flange is one of the weakest member in the Rotor shaft assembly of a Rotavator which will get fail i.e. there might be the chances of bending of Flange due to sudden impact force. So to avoid such failures it is required to change the material or increase the Yield strength of material, so we suggest EN 19 (oil Quenched And Drawn) Material for flange which is having better impact and load carrying capacities then AISI 1020.

IX. CONCLUSION

From analysis it is found that the flange is one of the weakest member in the Rotor shaft assembly of a Rotavator which will get fail i.e. there might be the chances of bending of Flange due to sudden impact force. So to avoid such failures it is required to change the material or increase the Yield strength of material, so we suggest EN 19 (oil Quenched And Drawn) Material for flange which is having better impact and load carrying capacities then AISI 1020 and so the following conclusions are done;

- The result shows that as the speed of Rotavator increase, leads to increase in the impact forces acting on the blade and thereby stresses are induced in the flange which leads to failure of Rotor shaft flange.
- The Existing Flange has shown the improved results with the change of material EN 19 at the different working speeds. Hence it is recommended to use EN 19 material instead of AISI 1020 which gives better results and having more strength and safely withstand the different working conditions.
- Using EN 19 as material of flange the fatigue life cycle is found increased considerably by 95675.33 cycles.

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