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Design and Fabrication of Indexing Set Up for Drill Jig

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Abstract - The scope of this work is to design and fabricate the indexing set up for actuator support component drill jig which is having holes at 90^0 apart. Jigs are mainly used for mass production and for interchangeable parts concept in the manufacturing of the indexing system. In this work main focus is given to the shaft which is to withstand the weight of the drill jig and the work piece while indexing. To achieve this, these components such as shaft, hydraulic jack were designed and manufactured.

The theoretical calculations of deflection and stress of the shaft to withstand the load of the drill jig and work piece were done and compared with Ansys results. The results of theoretical and Ansys average deflection and average stress error found are 0.0715 percent and 0.627 percent respectively. Comparison has also been done with manual and hydraulic indexing cycle times. The results of average percentage time saved are 71.94 percent by hydraulic indexing.

Key Words: drill jig, hydraulic jack, rotating shaft.

1. INTRODUCTION

This work includes only the indexing part and this indexing process includes many parts and are designed by using basic design formulae from the design data hand book and the main important part in this process is a shaft which is to take the load of the jig, job and the drilling force therefore the shaft is designed for the load acting and analysis is done for comparison of theoretical values and software values. Analysis is done by using ansys software.

1.1 STATEMENT OF THE PROBLEM

Indexing is to move from one set position to another in order to carry out a sequence of operation. In order to increase the production rate, reduce the labor effort and also to reduce the lead time, for all these purpose new indexing type of drill jig is used. The main object of reducing indexing time is achieved with the help of hydraulic jack which help the drill jig to be indexed by 180^0 for drilling holes on top and bottom face of the work piece. The rotation of the drill jig is obtained by the rotating shaft supported in two ball bearings.

For making this new indexing set up conventional machining is used and each part process is described in below chapters. This paper integrates all the journal papers on indexing mechanism and the knowledge of new indexing technique for drill jig.

1.2 METHODOLOGY

The basic methods used for the successful operation of the above concept are conventional machining, different methods of locating and clamping methods. Also the requirements of the project, such as increasing productivity, are reducing the lead time. Indexing is done with the help hydraulics. The setup is fabricated by using conventional machining operations. The parts such as hydraulic jack, shaft etc. are designed by using basic design formulae. Analysis is done by ANSYS software and modeling is done by SOLIDWORKS software.

2. DESIGNING THE PARTS OF INDEXING SET UP

2.1 DESIGN OF HYDRAULIS JACK

2.1.1 DESIGN OF PISTON ROD

This is a round in shape and it is having a length of 150mm. the piston rod is a mild steel SA36 GradeA. The ultimate stress of SA36 mild steel is 408MPa. The oil in the cylinder creates pressure in the barrel and the ram move in the barrel, the entire force acts on the piston. To design hydraulic barrel, it is required to calculate the load bearing capacity of the hydraulic barrel, which depends upon the load bearing capacity of piston.

2.1.2 CALCULATING THE LOAD BEARING CAPACITY OF PISTON ROD,

Assuming diameter of the piston rod is 15mm therefore we can find out the maximum load which it can withstand.

$$\text{Stress, } \sigma = \frac{load}{area} = \frac{F}{A}$$

Where, area $A = \pi r^2 \rightarrow$ r is the radius of piston rod

$$\text{Therefore } A = \pi \times (7.5)^2 = 176.7 mm^2$$

$$\text{Now, } F = \sigma \times A = 407.7 \times 176.7 = 72040 \text{ N}$$

If factor of safety of 4 is taken then, $F = \frac{407.7}{4} \times 176.7$

F = 18010.8 N =1801 Kg = 1.8 tons

This shows that the ram can withstand a maximum of 1.8 tons of load. But in this project it need to designed for 0.5 to 1 ton.

2.1.3 CALCULATING MAXIMUM INSIDE PRESSURE OF BARREL

Assuming 0.25 tons of load acts on piston ram. So, the pressure developed by the ram in the cylinder can be found by the following formules.

Prassure = F/A

Araa, $A = \pi \times r^2 = \pi \times (7.5)^2 = 176.7\ mm^2$

Force = 0.25 tons = 250 Kg = 250× 9.81 = 2452.5 N

Therefore pressure = 2452.5/176.7

$P = 13.879\ N/mm^2 = 13.879\ MPa$

This shows that pressure of 13.879 MPa will act on the walls of the cylinder when it is loaded with 0.25 tons.

2.1.4 DETERMINATION OF THICKNESS OF CYLINDER

For calculating thickness of barrel we are using Lame's equations from the design data hand book.

$$\sigma_r = \frac{b}{r2} - a \quad \text{.............} (1) \quad \rightarrow 12.1\ (DDHB)$$

$$\sigma_c = \frac{b}{r2} + a \quad \text{..........}(2) \quad \rightarrow 12.2\ (DDHB)$$

Where, σ_r= the radial stress

σ_c = the circumferential stress

a and b = constants

r = radius

As we are designing let us assume the internal diameter of barrel is 30mm. Than it is required to find out the OD of the cylinder.

Inner radous = $r_1 = \frac{Di}{2}$ = 30/2 =15mm

$\sigma_c$at inner radius r_i equal to 407.7 MPa

$$\sigma_c = 407.7 = \frac{b}{(15)^2} + a \quad \text{..........}(1a)$$

Since pressure at inner surface is 13.879 N/mm2

$$\sigma_r = 13.879 = \frac{b}{(15)^2} - a \quad \text{.........}(2a)$$

Adding (1a) and (2a)

$$407.7+13.879 = \frac{2b}{15^2} = 421.6 \times 15^2/2$$

b = 45866.25 N

Putting this value of b in (1) we get,

$$407.7 = \frac{45866.25}{15^2} + a$$

$a = 407.7 - 45866.25/15^2 = 206.5\ N/mm2$

a = 206.5N/mm2

therefore, lame`s equation for our case become

$$\sigma_r = \frac{45866.25}{r^2} - 206.5$$

Stress at the outer surface of cylinder should be zero,

i.e. $\sigma_r = 0$ (at radius = r_o)

$$0 = \frac{45866.25}{r^2} - 192 \rightarrow r_o^2 = \frac{45866.25}{192} = 239$$

r_o= 15.52mm

Outer diameter $d_o = 15.5 \times 2$ = 31mm

Therefore, thickness of the bareel (t) = outside radius – inside radius

$t = r_o - r_i$=31-30 = 1mm

2.1.5 BASE DESIGN

The base should be designed such that it must take the whole weight of the cylinder, ram, and other parts of it which is designed as below,

Area of the base on which load will acts is given as,

$A = a^2 - b^2$ .... (3)

Where a = size of the external face

b = size of the internal face

Assume, a = 40mm and b = 35mm

Then, wall thickness of the base is given as below

$$T = \frac{a-b}{2} = \frac{40-35}{2} = 2.5\text{mm}$$

Therefore, the Stress in the barrel for a load of 0.25 tones calculated as.

$$\sigma = \frac{force}{area} = \frac{9.81\times250}{40^2-35^2} = 6.54 \text{ N/mm}^2$$

From the above stress it shows that the stress is less than the maximum stress. Hence the design is safe.

2.2 SHAFT DESIGN

The shaft is a rotating member and is supported between the two bearings and it is subjected to bending as it is to with stand the load of the jig and the job such that the shaft must withstand the load applied without failure. Our aim is to design the shaft such as diameter of the shaft, maximum stress point on the shaft and bending moment such that it must be able to withstand the more than the actual load. The shaft with two supports and a point load is as shown in figure below,

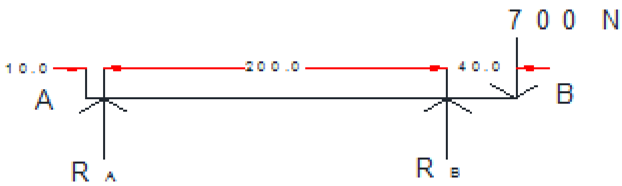

Figure 1 Support reactions on beam

From the above it shows that,

$$R_A + R_B = 700 \text{ .... (1)}$$

Taking the moment at point A, than

$$700 \times 0.25 = R_A \times 0.01 + R_B \times 0.21$$

$$175 = 0.01\, R_A + 0.21\, R_B \text{ .... (2)}$$

Solving equations 1 and 2 we get,

$R_A = -140$ N and $R_B = 840$ N

From the above two reactions Ra and Rb it is clear that the reaction force R_A should be in the downward direction.

The bending moment of the shaft can be calculated by using the SFD and BMD diagram as given below,

Figure 2 Shear force and bending moment diagram

From the above diagram it is clear that the maximum bending moment is at the point (3) and the maximum **B.M. is 28 Nm.**

W.K.T. The bending equation,

$$\frac{\sigma_b}{y} = \frac{M}{I} = \frac{E}{R} \qquad \sigma_b = \frac{M}{I} \times y \text{ ...... (a)}$$

We know that for mild steel material maximum bending stress is 407.7 N/mm^2 Taking the f.o.s (factor of safety) as 4 than bending stress $\sigma_b = 407.7/4 = 102$ N/mm^2.

Moment of inertia $I = \frac{\pi d^4}{64}$ and $y = d/2$

Now equation (a) changes to

$$102 \times 10^6 = \frac{28}{\frac{\pi d^4}{64}} \times \frac{d}{2} \quad \rightarrow \quad 102 \times 10^6 = \frac{32 \times 28}{\pi \times d^3}$$

$$d^3 = \frac{32\times28}{\pi\times102\times10^6}$$

Therefore, d = 14.5 mm.

As we are fabricating for safe taking two times the actual diameter as 30 mm.

3. 3D MODELLING OF PARTS OF INDEXING SETUP

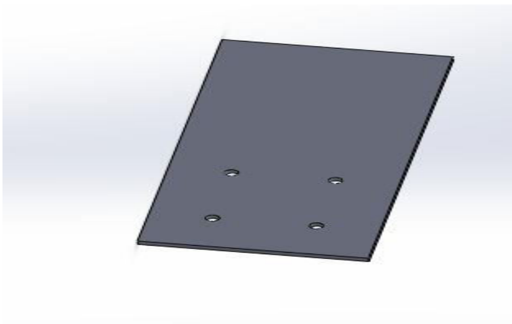

Figure 3: 3D model of base plate

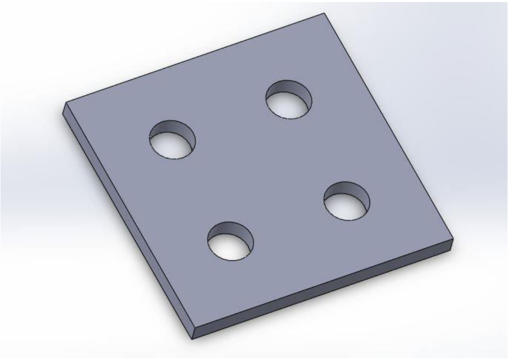

Figure 4: 3D model of middle plate

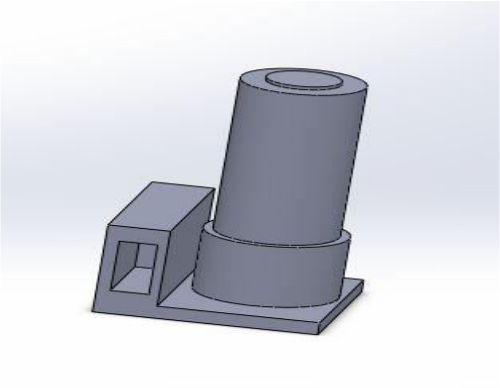

Figure 5: 3D model of hydraulic jack

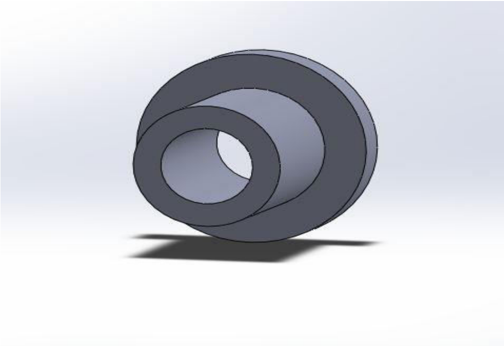

Figure 6: 3D model of bush

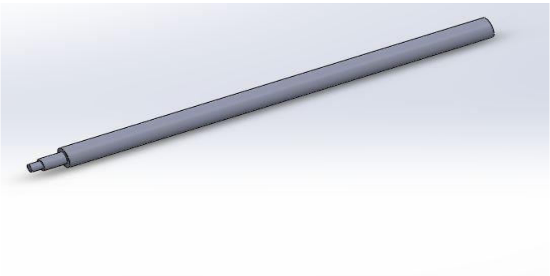

Figure 7: 3D model of guide rod

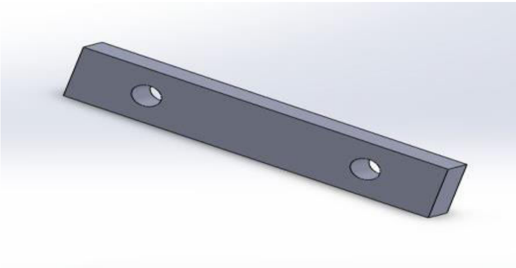

Figure 8: 3D model of guide rod brace plate

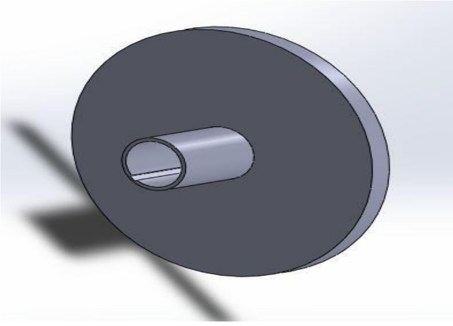

Figure 9: 3D model of flange plate

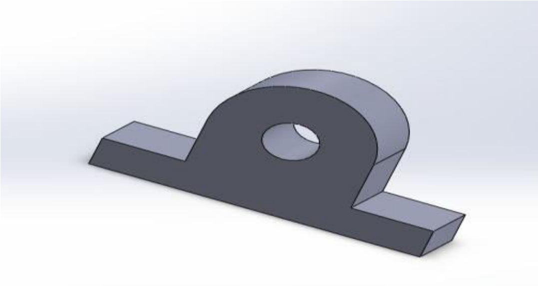

Figure 10: 3D model of bearing housing

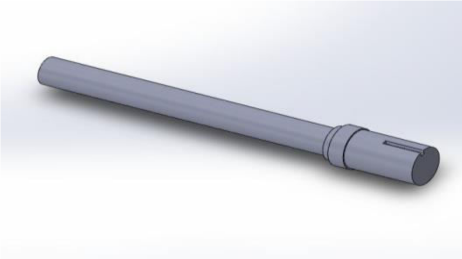

Figure 11: 3D model of shaft

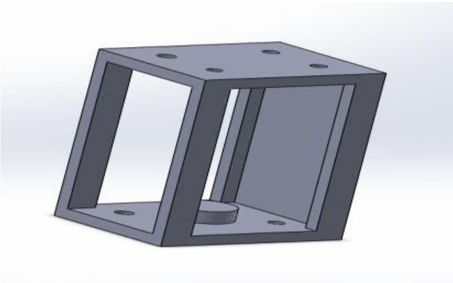

Figure 12: 3D model of drill jig

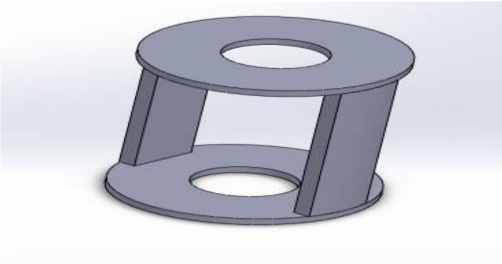

Figure 13: 3D model of work piece

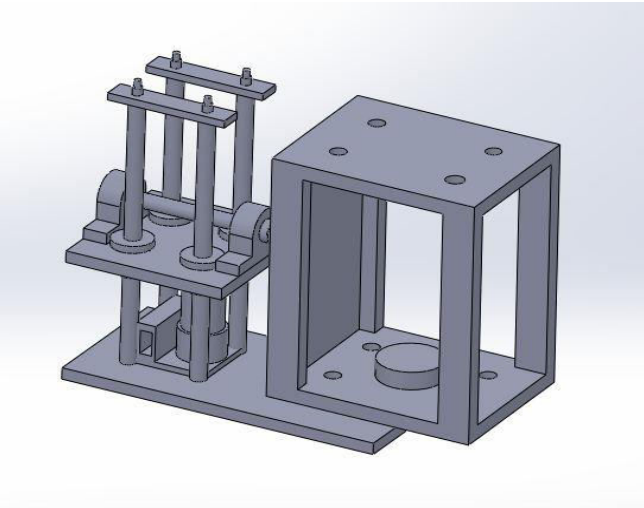

Figure 14 3D model of indexing set up assembly

4. EXPERIMENTAL PROCEDURE

Initially in the beginning of the operation the work piece is fixed in the drill jig after fixing the job the scale is checked and it is locked by a nut. In the beginning of operation the drill jig rests on the base plate and drilling is done to the job with the use of drilling machine. Now hydraulic jack valve is tighten and then with the help of the lever the jig is lifted and it is indexed to 180 degree. Than the jack valve is loosened than the jig is automatically comes down and rests on the base plate and then the valve is tighten. Again drilling to the other face is done on the drilling machine, while drilling the jig is fixed with the help of lock and pin arrangement and then the job is removed from the drill jig. Then new job is fixed and the process of operation is repeated.

5. MATERIAL PROCESS DETAILS

Table 1 Part process details

Name	Material	Quantity	Process
Base plate	Mild steel	1	Facing, drilling, milling
Middle plate	Mild steel	1	Facing, drilling, milling
Bush	Gun metal	4	**Facing drilling milling**
Guide rod	EN8	4	Facing, grinding, hardening
Fixing plate	EN8	2	Drilling
Shaft	EN8	1	Turning, grinding, key slotting
Circular plate	MILD STEEL	1	Facing , drilling, key slotting

6. SHAFT ANALYSIS BY ANSYS SOFTWARE

The shaft which is to withstand the load of the drill jig and the work piece such that it can easily withstand the load acting on it without failure for that we have to analyze the shaft for the load acting.

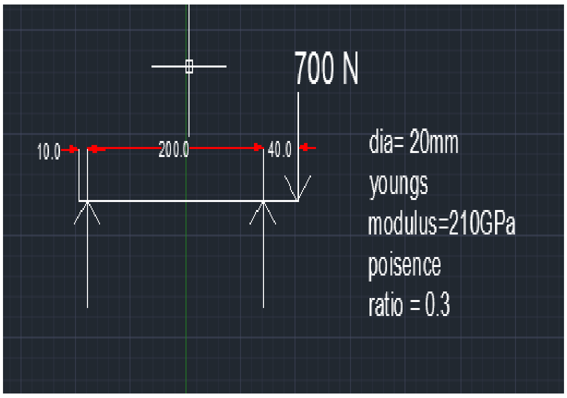

Figure 15: shaft description

In ANSYS firstly the deflection of the beam or shaft is found and also the stresses generating in the beam due to the applied load are found i.e. the values of maximum stress and minimum stress are calculated and then the ANSYS values are compared with the values of the theoretical calculations.

6.1 DEFLECTION OF BEAM IN ANSYS AS SHOWN BELOW,

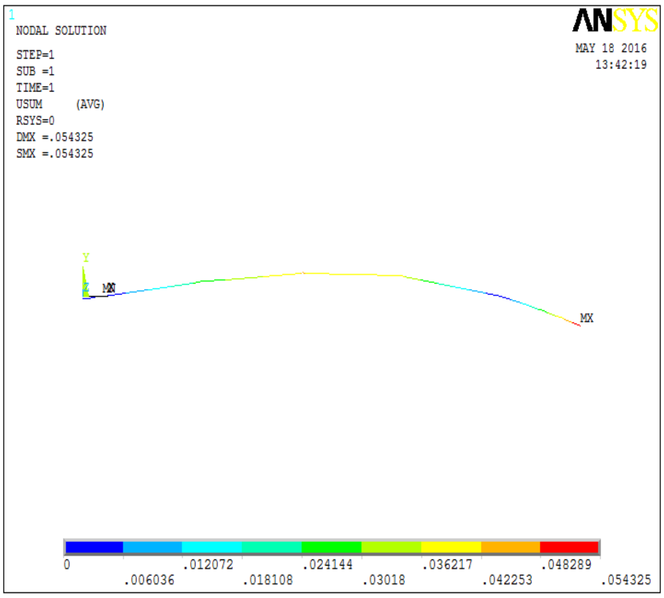

Figure 16: deflection of beam

From figure 16 it is clear that the maximum deflection is at the right end of the beam having the value 0.054325mm which is very less so the beam is safe as the deflection is concerned.

6.2 MAXIMUM AND MINIMUM STRESS OF BEAM

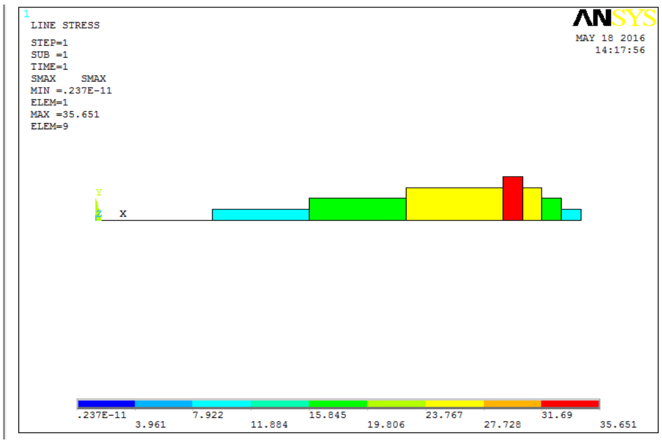

Figure 17: max and min stress of beam

Above figure shows that the maximum stress is at the point where the color has become red and is having the maximum value 35.651 N/mm^{2}and the minimum stress is occurring at the left end of the beam showing the blue color as shown in figure. From the above it is clear that the maximum stress 35.651 N/mm^2 is less than the maximum bending stress of the beam. Hence this shows that the beam is under safe.

7. COMPARISON OF ANSYS AND THEORETICAL VALUES

Table No 2

Title		Theoretical value	Ansys value	Error
Deflection	Maximum	0.05436mm	0.0543mm	0.11%
	Minimum	0.00638mm	0.006036mm	0.033%
Avg max and min deflection error				0.0715%
stress	Maximum	35.653mpa	35.651mpa	0.0056%
	Minimum	$0.233*10^{-11}$mpa	$0.237*10^{-11}$mpa	1.25%
Avg max and min stress error				0.627%

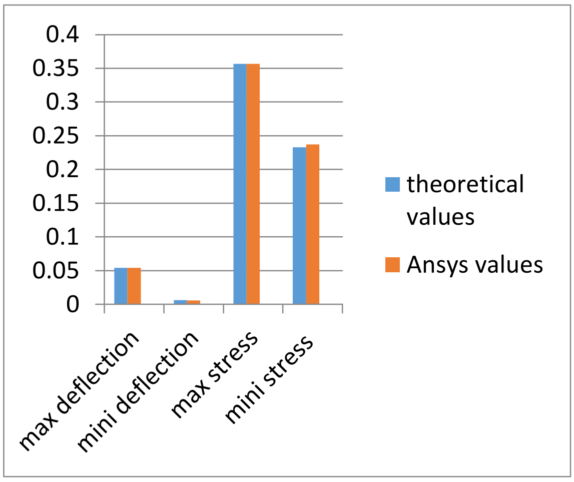

Figure 18. Bar chart of comparison of max and min theoretical, Ansys values

8. EXPERIMENTATION OF MANUAL INDEXING AND WITH HYDRAULIC INDEXING

Table No 3

Job No	Manual indexing		Indexing by hydraulic jack		Avg %age Time saved
	Cycle time in minutes	Indexing time sec	Cycle time in minutes	Indexing time in sec	
1	30	30	30	10	66%
2	30	32	30	10	68.7%
3	30	32	30	10	68.7%
4	30	34	30	10	70.5%
5	30	34	30	10	70.5%
6	30	38	30	10	73.6%
7	30	38	30	10	73.6%
8	30	40	30	10	75%
9	30	42	30	10	76.1%
10	30	42	30	10	76.1%
Avg %age indexing time saved					71.9%

Average manual cycle time = 30 minutes

Average manual indexing time = 36.8 seconds

Therefore,

Average manual time = 30+36.8 =30.368 minutes

Average cycle time with jack = 30

Average indexing time with jack = 10 seconds

Average indexing time = 30+10 = 30.10 seconds

We know that,

Manufacturing lead time MLT = n_o (Tsu + Q x T_c + T_{no}) … (11)

Where, n_o = number of operation on work piece = 2

T_{su} = set up time in manual operation = 60 minutes

Q = number of work pieces = 10

T_c = cycle time of manual operation = 30.36 minutes

Tno = non operation time in manual operation = 40 minutes

Equation 1 gives,

MLT = 2 (60 + 10 x 30.36 + 40) = 807.2 minutes

For 8hr/day

MLT with manual = $\frac{807.2}{8\times60}$ = 1.68 days x 24hrs/day = 40.32 hrs.

MLT with manual indexing = 40.32 hours ……… (a)

Now calculating MLT of indexing with hydraulic jack,

Average indexing cycle time Tc = 30.10 minutes

Set up time in the use of jack Tsu = 10 minutes

Therefore,

MLT with the hydraulic jack = 2 (10 + 10 x 30.10 + 10)

= 642 minutes

For 8hr/day,

MLT with hydraulic = $\frac{642}{8 X 60}$= 1.33 days x 24 hrs/day = 32 hours

MLT with hydraulic = 32 hours. ……… (b)

from the above a and b values it shows that with manual indexing time required to produce 10 number of pieces requires 41 hours and with the use of hydraulic jack indexing it takes 32 hours for 10 pieces.

So this proves that, this concept chosen for indexing helps to improve the production rate and operation time.

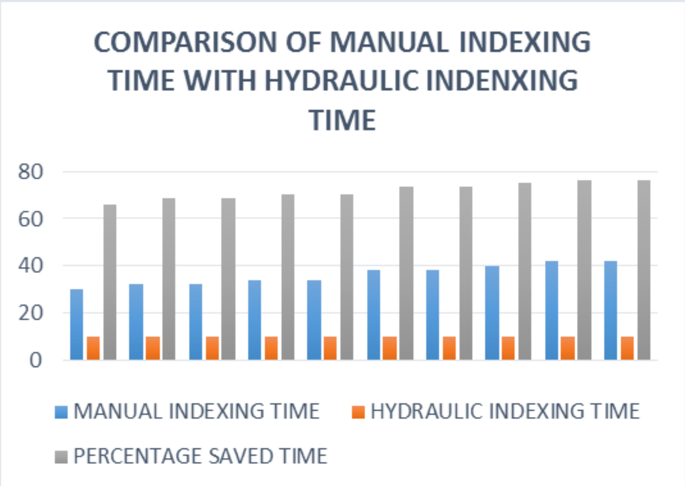

Figure 19. Bar chart of comparison of manual indexing, hydraulic indexing time and %age time saved

9. Discussion of results

- Deflection of the beam in the ansys is maximum at the right end of the beam i.e. 0.0543 mm, it is very less so the beam under load is in the safe limit.
- The maximum stress in the beam is 35.651 N/mm^2 is lesser than the ultimate value of mild steel material so the beam is safe.
- From the above ansys results and theoretical values it is clear that our design of beam is under the limit and it is safe for the indexing mechanism operation.
- Production rate with the use of hydraulic jack is more compared to the manual indexing.
- Time required for production of work pieces is lesser with the use of hydraulic jack than manual.
- By this project concept 22% of the production rate is increased.

10. Conclusion

1. The new indexing type of drill jig has been successfully designed and developed as per the needs of the company from the existing model.
2. This project can be the perfect solution for the problem that was occurred.
3. With the use of this indexing mechanism in industry will certainly help in increasing production rate, production time and also reducing the production cost.
4. The production rate is increased by 22% when compared to the previous operation.
5. The process of operation is easier by this concept compared to manual indexing

11. Scope of the project

1. This concept can be used in small scale industries for increasing production rate and to minimize the production time.
2. By replacing hydraulic jack with power pack still the production time is reduced.
3. By this new indexing set up, jigs of different sizes can be used in this for drilling operation.

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