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Development of a Mechanical Puller

Emagbetere Eyere
Lecturer Department of Mechanical
Federal University of Petroleum Resources, Effurun

Oreko, Benjamin Ufuoma
Lecturer Department of Mechanical
Federal University of Petroleum Resources, Effurun

Agberegha, Orobome Larry
Lecturer Department of Mechanical
Federal University of Petroleum Resources, Effurun

Oghenekowho Peter
Lecturer Department of Mechanical
Federal University of Petroleum Resources, Effurun

Abstarct:- In this work, a simple, low cost, less weight and less energy to operate mechanical puller was developed. The puller screw shaft was first design for, followed by the nut and then its collar top. Non-conventional shapes were assumed for the collars and handles. Finite element analyses were used to evaluate the von-misses stress in the arm and collar which had complex shapes. The result of finite element analysis revealed that the puller is not likely to fail under the design working conditions. The designed 3 arm mechanical bearing puller was fabricated and was successfully used to remove a number of bearings stuck in shaft.

***Keywords**:- Mechanical puller, Finite Element Analysis, Shaft, Puller screw, von-misses stress.*

I. INTRODUCTION

In the routine or corrective maintenance of rotating systems, a fast, reliable and non-destructive approach need to beadopted. Use of hammerand drift pins to removebearings, gears, pulleys and other components stuck in shafts can damage the shaft or the component.Furthermore, such method of removing stuck components can be unsafe and is a form of occupational hazard. While occupational hazards have been a concern for long (1), harmful habits have reportedly led to injuries and other work related health issues in the automobile industry (2). A puller,also called an extractor, can sufficiently be used to safely remove components stuck in a shaft.

Development of puller for removal of components from shaft started almost a century ago (3). There are two types of pullers namely;the hydraulic and the mechanical types.The hydraulic pullers work base on hydraulic principle. They are easier and quicker to use, also they are suitably used when very largeextraction forces are required (4).Suryawanshiet. al. (5)reported the design and fabrication of a hydraulic puller. Hydraulic pullersare complicated and costly to acquire by local technologyin developing countries.

The mechanical puller is a simple device in that it has few and easy to construct components, namely; the screw, a bolt/head, linkage arm (jaw) and a handle. The jaws are either two or three. Currently, there exist different jaw designs (6).The mechanical pullers work base on the principle of power screw,that is, they change angular motion into linear motion totransmit power.The principle of operation and design details of power screw are discussed in the literature (7; 8; 9). Abella*et. al.*(10) reported the design of an electrically controlled bearing puller. They however focused on the electrical motor. Although, much have been reported on the analysis of power screws(11; 12), there is dearth of information on stress analysis of other parts of a puller.

The aim of this work is to develop a mechanical puller that can easily be fabricated using available local technology and investigate the stresses acting on its parts using finite element method.

II. DESIGN AND ANALYSIS

➢ *Materials Selection*

In line with (13; 14; 15),an investigation of the design requirement preceded the selection of materials for the project. The major design considerations were; high yield Strength, low mass, resistance to abrasive wear, resistance to buckling, availability, low cost and ease of manufacture.

Material selection was based on their characteristics, properties and suitability for the operational condition. The approach was to identify the connection between functional requirements and the material properties.The parts of the puller, their functionsandthe respective materials selected are presented as follows:

A. linkage (arm)

The linkages work together to ensure that the rotational motion of the screw is converted to linear motion to pull the bearing out from the shaft. The purpose of the arms is to support the puller and enable it to withstand compressive load exerted on it.

The link is a bit complex and thus requires casting as a manufacturing process. For this reason, gray cast iron as a material is selected for the frame. Additionally, cast iron is cheap, has high compressive strength and can be used to manufacture any complex shape without involving costly machining operations.

B. Screw

The screw is to transmit power while converting rotational motion into translational. It is to withstand torsional moment, compressive force and bending moment. The square type screw profile was selected for this work due to its higher efficiency and self-locking and ease of fabrication. Screws are usually made of steel where great resistance to weather or corrosion is required. The material selected is plain carbon steel because of its excellent workability, wide range of attainable combinations of strength properties and its availability.

C. Nut

There exists a relative motion between the screw and the nut which causes friction, friction in turn causes wear of the material used for screw and nut. It was therefore necessary to select a softer material for the nut. As a result, phosphor bronze which is a copper alloy with small percentage of lead was selected for the nut. Its advantages are: good corrosion resistance, low coefficient of friction, high tensile strength.

III. DESIGN AND MODELLING

A Design of Screw Shaft Spindle

Material specification selected for the screw shaft is plain carbon steel, BS 970 080M30, Hardened and Tempered, with yield strength of 200 MPa both in tension and pure compression and 190 MPa in shear.

- *Assumptions*

Maximum weight of the load to be pulled W = mg = 4kN
Tensional stress of the screw spindle material $\sigma_t = 200\ MPa$
Compressional stress $\sigma_c = 190\ MPa$
Factor of safety = 4
Implying that working stress $\sigma_w = 50\ MPa$
Coefficient of friction between screw and nut = 0.15

- *Core Diameter*

The core diameter is determined by considering the screw to be under pure compression. That is;

$$\mathrm{W} = \sigma_\mathrm{w} \times \mathrm{A_c}$$

Where
A_c = Cross sectional area of the screw shaft = $\pi/4\ (d_c)^2$
d_c= Core diameter

$$\mathrm{W} = \sigma_\mathrm{w} \times \pi/4(d_c)^2$$
$$4000 = 50 \times \pi/4(d_c)^2$$

d_c= √4000/39.5 = 101.26

$$d_c = \sqrt{101.26}$$
$$d_c = 10\ mm$$

Taking factor of safety $f.s$ = 4

This value of determined core diameter informs the selection of core diameter d_c of 16 mm which is the closest available dimension.For square threads of fine series, the following dimensions of screw are selected based on literature(8): The core diameter d_c=16mm, d_o=18mm and pitch p=l = 2mm.

- *Torque Required to Rotate the Screw*

This torque is equivalent to the torque required to lower load by square threaded screws

$$T_1 = P * \frac{d}{2} = Wtan(\phi - \alpha)\frac{d}{2}$$

Where P = effort applied at the circumference of the screw to lift the load

d=mean diameter of the screw

W=load to be lifted

α=helix angle

ϕ=friction angle

We know that torque required to rotate the screw is the same torque required to lift the load which is given by;
d_m= (d_o+d_c)/ 2= (18+16)/2=17mm

And tanα= p/πd_m=2/2π×17=0.03745

$$\alpha = tan^{-1}0.0375$$
$$\alpha = 2.15$$

Assuming coefficient of friction between screw and nut, μ=tanθ=0.15

$$\theta = 8.53$$

Then $T1$=[4000 tan (2.15+8.53)]×8.5 =6412.07Nm

- *Screw Stresses*

Compressive Stress due to axial load using the new core diameter is,

$$\tau = \frac{16T}{\pi(d_c)^2}$$
$$\tau = \frac{16 \times 6426}{\pi(16)^2}$$
$$\tau = 7.99N/mm^2$$

(isτstanding for compressive stress?)

- *Principal Stresses*

Maximum principal stress is as follows:

$$\sigma_{c\ (max)} = \frac{1}{2}\left[\sigma_c + \sqrt{\sigma_c{}^2 + 4\tau^2}\right]$$
$$\sigma_{c\ (max)} = \frac{1}{2}\left[19.9 + \sqrt{19.9^2 + 4 \times 7.99^2}\right]$$
$$\sigma_{c\ (max)} = 22.71N/mm^2$$

- *The maximum shear stress in screw shaft spindle*

$$\tau_{\ (max)} = \frac{1}{2}\left[\sqrt{\sigma_c{}^2 + 4\tau^2}\right]$$
$$\tau_{\ (max)} = \frac{1}{2}\left[\sqrt{19.9^2 + 4 \times 7.99^2}\right]$$
$$\tau_{\ (max)} = 12.76\ N/mm^2$$

Note that material shear stress $= \tau = \frac{\tau_{ec}}{F.S} = \frac{120}{4} = 30\ N/mm^2$

Check: These maximum shear and compressive stresses are less than the permissible stresses, hence the spindle or shaft is safe.

2.2.2 Design for Nut

- *Height of the Nut*

We find the height of the nut (h) by considering the bearing pressure P_b on the nut. The bearing pressure on the nut is given by;

$$P_b = \frac{W}{\frac{\pi}{4}[(do)^2 - (dc)^2]n}$$
$$18 = \frac{4000}{\frac{\pi}{4}[(18)^2 - (16)^2]n}$$

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Assuming the load is uniformly distributed over the entire cross section of the nut and substituting for the known values we get the number of threads in contact,

$$n = \frac{4000}{961.2} = 4.16 \quad say\ 6mm$$

Where

n = Number of threads in contact with screwed spindle

Material specification for the nut is phosphor bronze which has tensile stress = 150MPa, compressive stress = 125MPa, shear stress = 105MPa, safe bearing pressure not exceeding 17MPa and a coefficient of friction of 0.1.

Then height of the nut is as follows;

$h = n \times p$

$h = 6 \times 2 = 12\ mm$

Butfor the sake of construction, the height of the nut was 40mm

Check: For a safe nut height $h \leq 4d_c = 64m(8)$

- *Stresses in the Screw and Nut*

Shear stress in the screw is as follows;

$$\tau(screw) = W/\pi n.d_c.t$$
$$\tau(screw) = 4000/\pi \times 6 \times 16 \times 1$$
$$\tau(screw) = 13.3N/mm^2$$

Where $t = Thicknessofscrew = p/2 = 1mm$

And shear stress in the nut is as follows;

$$\tau(nut) = W/\pi n.do.t$$
$$\tau(nut) = 4000/\pi \times 6 \times 18 \times 1$$
$$\tau(nut) = 11.79N/mm^2$$

Where $t = Thicknessofscrew = p/2 = 1mm$

Check: These stresses are within permissible limit; hence, design for the nut is safe.

- *The outer diameter of Nut*

Outer diameter $D1$ is found by considering the tearing strength of the nut.

$W = \pi/4[(D1)^2 - (do)^2]\ \sigma_t$

Where $\sigma_t = Tearingstrengthofthenut = Tensilestress$

$\sigma t = 95MPa$

Then we get $D1$ as follows;

$D_1 = 19.36mm$, Say $D1 = 20mm$

- *The outside diameter of Collar*

Outside diameter $D2$ is found by considering the crushing strength of the nut collar.

$$W = \pi/4[(D2)2-(D1)2]\ \sigma_c$$

Where $\sigma_c = Crushingstrengthofthenut = Compressivestress$

$$\sigma_c = 85N/mm^3$$

Then we get $D2$ as follows; $25 = 24132.60\pi 4[(D2)2-(37)2]$

$$D2 = 21.3mm, \text{ Say } D2 = 22mm$$

- *Thickness of the Nut of Collar*

The thickness of nut collar $\boldsymbol{t1}$ is found by considering the shearing strength of the nut collar.

$t1 = W/\pi D1.\tau$

Shearing strength of nut collar =40N/mm^2

Therefore $t1 = 4000/\pi \times 22 \times 40$

$t_1 = 1.644mm$,

Therefore $t_1 = 2mm$

2.2.3 Efficiency of the bearing puller

Efficiency of the puller is given as follows:

$$\eta = \frac{\text{Torque required to rotate screw with no friction}}{\text{Total torque output}} = \frac{T_o}{T}$$

$$\eta = \frac{T_o}{T}$$

But $T_o = W\tan\alpha \times d_m/2$

$T_o = 4000 \times 0.0375 \times 17/2$

$T_o = 1275$N/mm

$$\eta = \frac{1200}{6424} = 0.1984 \text{ or } 19.84\%$$

B Finite Element Analysis

Finite element analysis was conducted on the trust collar, with solidworks simulation expressby considering the Von Mises theory, the material yield strength of the trust collar which bears most of the loads. The collar and puller arm were both subjected to finite element analysis:

Orthographic projection of the developed puller collar, arm and connector are shown in Figures 1-3 respectively.The 3D model for the puller produced using Solidworks are shown in Figures 4 and 5.The values of reaction forces applied to the model are shown in Table 1. Table 2 contains information of the model. The linear elastic isotropic model was adopted for the analysis, maximum von-misesfailure criterion was selected and the respective material types were chosen from the material library. Figures 6 and 7 are the mesh models of the collar and handle respectively. The respective mesh parameters are shown in Table 3.

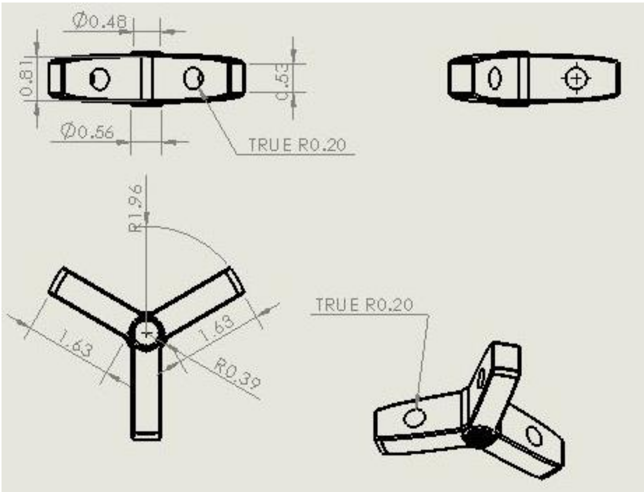

Fig 1:- Orthographic projection of the puller collar

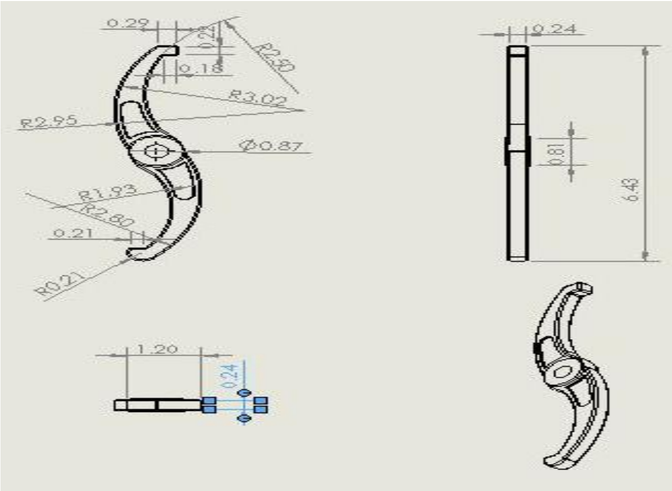

Fig 2:- Orthographic projection of the puller arm

Fig 5:- 3D model of the puller arm/handle under load condition

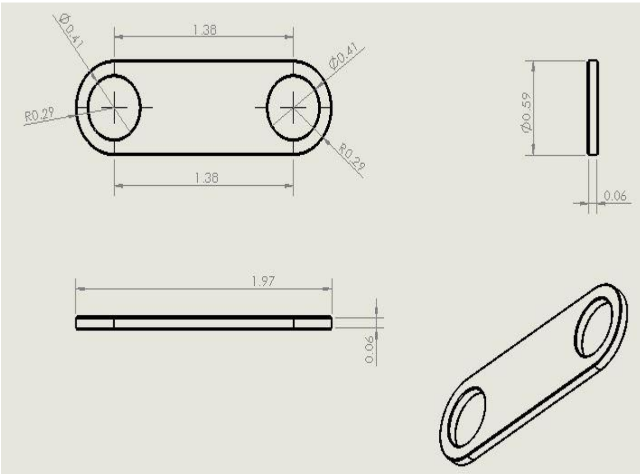

Fig 3:- Orthographic projection of the puller connector

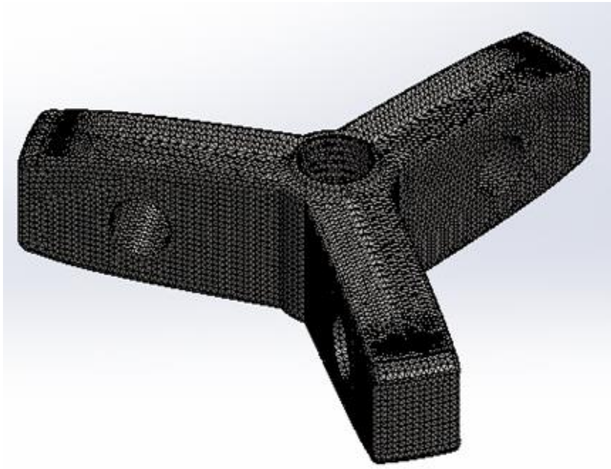

Fig 6:- Mesh model of the puller collar

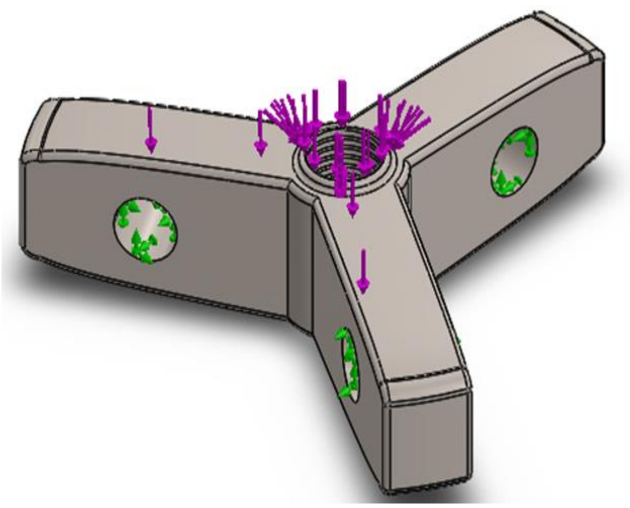

Fig 4:- 3D model of the puller collar under load condition.

Fig 7:- Mesh model of the Puller handle

Components	X	Y	Z	Resultant
Collar	-0.35358	355.997	0.765008	355.998
Arm / handle	0.004056	99.9884	-0.00595	99.9884

Table 1. Reaction forces acting on the models

Parameter	Collar	Arm/handle
Mass (kg)	0.280783	0.136219
Volume (m^3)	3.65E-05	1.77E-05
Density (kg/m^3)	7700	7700
Weight (N)	2.75167	1.33495

Table 2. Model information

	Collar	Arm/handle
Total Nodes	351677	9782
Total Elements	244184	5701
Maximum Aspect Ratio	19.977	33.165
% of elements with Aspect Ratio < 3	97.9	77.8
% of elements with Aspect Ratio > 10	0.0156	1.37
% of distorted elements (Jacobian)	0	0
Time to complete mesh(hh;mm;ss)	0.000532	6.944E-05

Table 3. Mesh Information

C Fabrication

Table 4. is a summary of the techniques with which the different parts of the puller were fabricated. The fabricated puller is shown in Figure 8. The fabricated puller was successfully used to remove a number of bearings stuck in a shaft.

Component	Quantity	material	Means of forming
Linkage arm	3	Alloy steel	Casting
Collar	1	Cast iron	Casting
Screw shaft	1	steel	Cutting and machining
Bolt	6	Alloy steel	Casting and machining
Washer	6	Alloy steel	Casting

Table 4. Selected materials for different components

Fig 8:- The constructed Puller

IV. FINITE ELEMENT ANALYSIS RESULTS

Figure 9 shows the color plot solution for finite element analysis of the puller collar. It observed that the value of von-mises stress is higher than the working stress. This is an indication that the device will satisfactorily perform its function. The highest stress values were at the threaded areas where friction is overcome in combination with the puller screw. Figure 10is the color plot solution for finite element analysis of the puller arm. It was found that the von-misses stress were higher than the working stress of the designed puller. The values of stress were highest for the thinner part of the puller arms. The arm is not likely to fail under normal working conditions as its maximum Von-mises stress is higher than the working stresses.

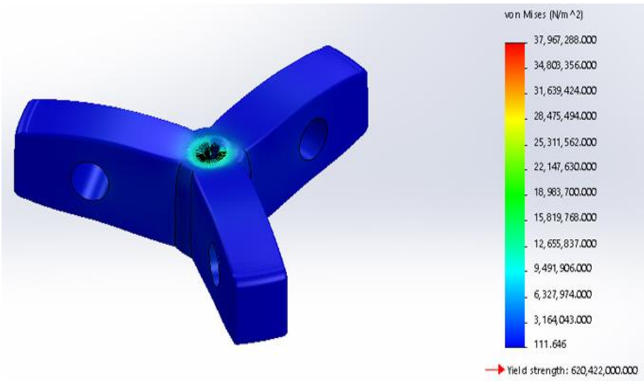

Fig 9:- Finite element analysis solution of the Puller collar

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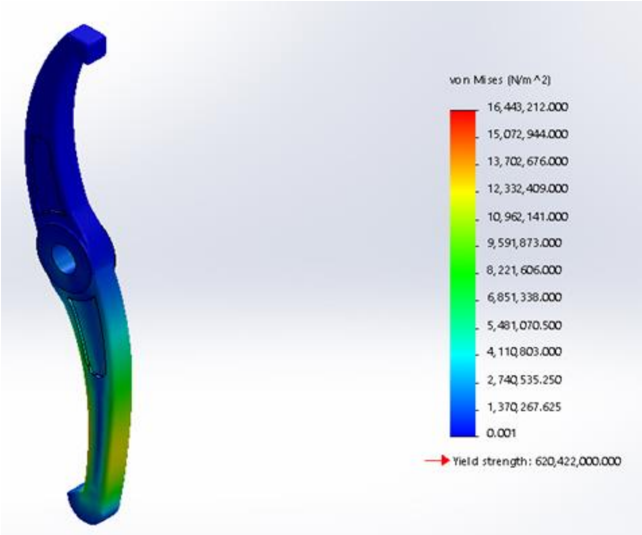

Fig 10:- Finite element analysis solution of the puller handle

V. CONCLUSION

A 3 arms mechanical bearing puller was designed, and stresses on its part with irregular shape were analysed using finite element method. The fabricated model prototype was tested by using it to remove bearing stock on a shaft, the resulting performance was satisfactory. Finite Element Analysis of the puller arm and collar showed that the puller is not likely to fail under its designed working conditions.

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