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A Review on Design and Analysis of Hydraulic Arm for Scrap Lifting

Sagar Dhotare[1], S.S.Chavan[2], Pratik.D.Mhatre[2], Ashok.G.Jadhav[2], R.B.Choudhary[2]

ABSTRACT: Industrial Material Handling is an important phenomenon and it involves handling, movement, storage, and control of material or equipment's through the entire procedure of manufacturing. As a process material handling incorporates a wide range of manual, semi-automated, tools and equipment's. One of the aim of this project is to design an hydraulic arm for lifting of scrap material which would result in reduction of material handling time. Design process includes developing material handling equipment on CAD software as per the company's requirements. Project also includes the theoretical and software based analysis. It was understood that how different parameters affect an hydraulic arm's lifting capacity and maximum reach. The hydraulic arm was tested in analysis software and the results indicates that it can perform the lifting task properly.

Keywords: Material Handling, Scrap Lifting, Hydraulic Arm, Semi-Automated.

1. INTRODUCTION:

In order to convert the raw materials into finished products, it is essentials that one of the three basic elements of production, i.e. material, men or machines should move. In majority processes, it is the material that moves from raw material stage to the stage of when it becomes the finished good. Materials handling occurs whenever a material is moved may be in a manufacturing, distribution (warehouse), or office environment. Materials handling also occur during preparation for shipment, transportation may be by sea, air or land, and moving material in and out of carriers.

Material handling embraces the basic operations in connection with the movement bulk, packaged and individual products in semi-solid or solid state by means of gravity manually or powers-actuated equipment and within the limits of individual producing, fabricating, processing or service establishment. Material handling amounts to 15 to 25% of the total cost of the products.

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Sagar Dhotare, S..S.Chavan, Pratik.D.Mhatre, Ashok.G.Jadhav2, R.B.Choudhary
ashokagjadhav@gmail.com

Department of Mechanical Engineering, Vishwaniketan's iMEET, Khalapur-410202, India

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The essential requirements of a good materials handling system may be summarized as:

(i) Efficient and safe movement of materials to the desired place.
(ii) Timely movement of the materials when needed.
(iii) Supply of materials at the desired rate.
(iv) Storing of materials utilizing minimum space.
(v) Lowest cost solution to the materials handling activities

2. LITERATURE REVIEW:

Mr. Altaf S. Shaikh and Dr. B.M. Shinde [1] In this paper Finite element analysis (FEA) of existing excavator arm is compared with optimized arm for stresses and deflection. This paper discuss about finite element based optimization of excavator arm and thus helped in finding out the most appropriate design of which a prototype is fabricated and tested.

Ramesh kumar and R.jaison [2] This paper consist of designing of a detachable backhoe and loader components that is to be fitted on a agricultural tractor to lift a load of 2000N.

The detachable type backhoe components are designed using theoretical calculations and with modelling module using CREO Parametric.ANSYS WORKBENCH are used as a tool to redesign the component without making the prototype and the loading condition are simulated and to make the necessary changes at the design level.

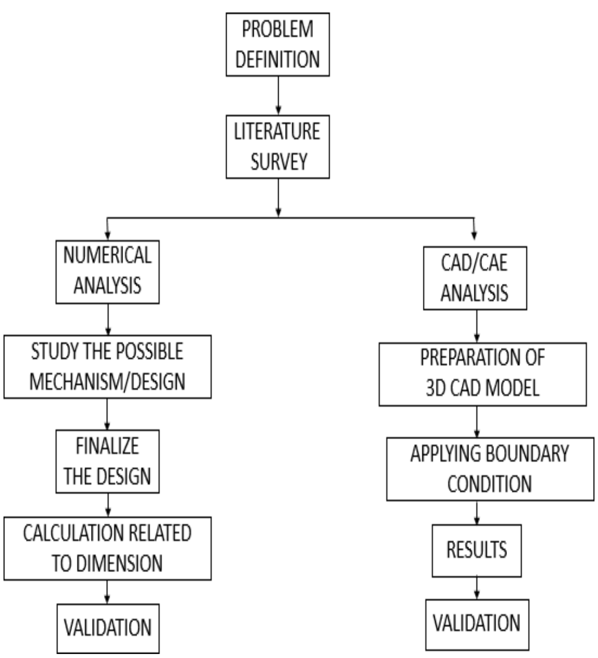
Shilpa D. Chumbale, Prasad P. Mahajan[3] In this new mechanism of excavator arm is designed and the Pro-e software is used for making the 3D model of the excavator arm linkage. By using ANSYS workbench software static analysis of each of the excavator arm component is done at existing digging force. The

3. PROBLEM STATEMENT:

Material handling of raw materials is plays an important role in process of manufacturing as working speed of material handing will affect the production rate. In steel industries different types of material handling equipment's are used for raw materials. Respective company is using manual man work method for material handling of non-raw materials to respective area.

It has been observed by company that the method which is using is very time consuming. So, to reduce the time, and increase the efficiency of work, company is decided to design hydraulic arm for lifting raw materials scrap. Company needed the two separate designs for scrap unloading from trucks and for loading scrap materials into Bell Furnace with respect to the dimensions of plants.

3. METHODOLOGY:



4. DESIGN :

<u>FOR DESIGN 1</u>

A) Pin for Base Piston and Cylinder (Pin 11)

Oil –HP Enklo 68

Effective Length (le) = 100 mm

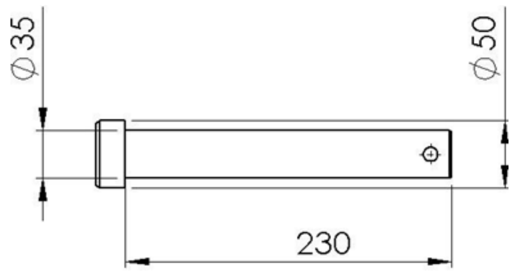


Fig. 1 Design of Pin11.

Pin Material-Hardened Steel 12.9 Operation

Pressure = 60 to 100 Bar

= 6 to 10 MPa Taking

Pressure(P)= 10 MPa

Pressure Force On Pin (F),

= P* Area of Piston (1)

$= [P * \frac{\pi}{4} * (d_p)^2] * 50\%$ Overload

$= [10 * \frac{\pi}{4} * (60)^2] * 1.5$

F = 42411.495 N

i. Checking for Shear Failure :

Allowable Shear Stress (τallowable) = 150 MPa

Area $= 2 * \pi/4 * d^2$ (2)

$= 2 * \pi/4 * 35^2$

Area = 1924.2255 mm²

$\tau_{indu} = F/A$ (3)

= 42411.5 / 1924.22

$\boldsymbol{\tau_{indu} = 22.040 < \tau_{allowable}}$

ii. Checking for Bending :

Allowable Bending Stress $(\sigma_{b\ allowable})$ = 300 MPa

UDL on pin (w) $= F / l_e$ (4)

= 42411.5 / 100

w = 424.11 N/mm

Maximum Bending Moment(M) $= [w * (l_e)^2] / 8$ (5)

$= [424.11*(100^2)]/8$

M = 530143.687 Nmm

Moment Of Inertia(I) $= [\pi/64] * d^4$ (6)

$= [\pi/64] * (35)^4$

I = 73661.757 mm⁴

Maximum Bending Stress,

$(\sigma_{bmax}) = [M * y] / I$ (7)

= [530143.6*17.5]/73661

$\boldsymbol{\sigma_{bmax} = 125.948\ N/mm^2}$

$\boldsymbol{\sigma_{bmax} < \sigma_{b\ allowable}}$

B. **Arm :**

Material- IS2062($\sigma_{t\ allowable}$=102.5 N/mm²)

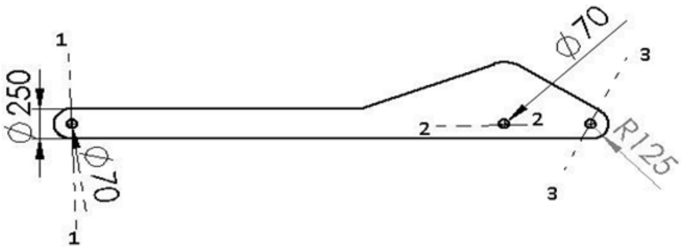


Fig. 2 Design of ARM

Section 2-2

Load (Q) = 33335.924 N
Diameter of Hole (d) = 70 mm
Width subjected to failure(B) = 3.5*d= 245 mm
Thickness (t) =20 mm

Checking for tensile failure

$$\sigma_t = P/A \tag{8}$$
$$= [Q/2] / [(B-d)*t]$$
$$= [16667.9/2] / [(245-70)*20]$$
$$\boldsymbol{\sigma_t = 4.76\ N/mm^2 < \sigma_{t\ allowable}}$$

C. Boom :

Material- IS2062($\sigma_{t\ allowable}$=102.5 N/mm²)

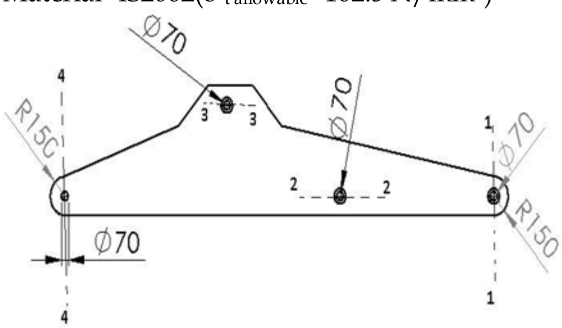


Fig.3 Design of Boom Arm

Section 3-3

Load (Q) = 68442.24 N

Diameter of Hole (d) = 70 mm

Width subjected to failure(B) = 3.5*d= 245 mm
Thickness (t) =20 mm

Checking for tensile failure

$$\sigma_t = P/A$$
$$= [Q/2] / [(B-d)*t]$$
$$= [34221.12] / [(245-70)*20]$$
$$\boldsymbol{\sigma_t = 9.77\ N/mm^2 < \sigma_{t\ allowable}}$$

D. Base :

Material- IS2062($\sigma_{t\ allowable}$=102.5 N/mm²)

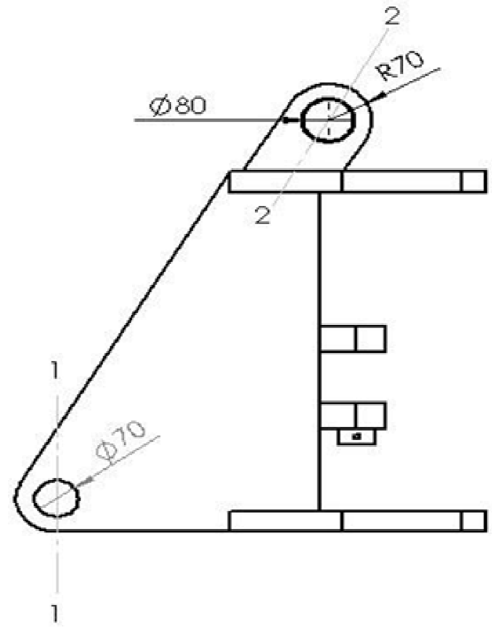


Fig. 4 Design of Base

Section 4-4

Load (Q) = 83152.8 N
Width subjected to failure (B) = 120 mm
Diameter of Hole (d) = 70 mm
Thickness (t) =40 mm

Checking for tensile failure

$$\sigma_t = P/A$$
$$= [Q/2] / [(B-d)*t]$$
$$= [41576.4] / [(120-70)*40]$$
$$\boldsymbol{\sigma_t = 20.77\ N/mm^2 < \sigma_{t\ allowable}}$$

FOR DESIGN 2

A. Pin for connecting moving and fixed base (lower) (Pin 10)

Effective Length (l_e) = mm

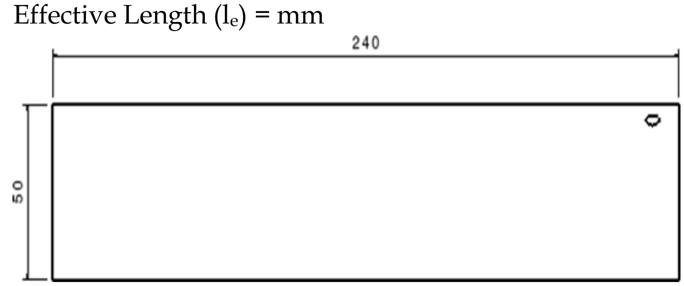


Fig. 5 Design of Base Plate

Total Force On Pin(F) = [Pressure Force (1,2) + Arm Wt.+Wt. Of Bush(2,3,4,5,6,7,8,9) + Wt. Of Pin (1,2,3,4,5,6,7,8,9) +Jaw Wt.+ Scrap Wt. + Connecting Rod Wt.+ Wt. of boom+Wt. Of Moving base+Wt of Cylinder

(1,2) + Wt. Of Piston (1,2)]

= 7633.519 Kg

F = 76335.19 N

i. Checking for Shear Failure

Allowable Shear Stress ($\tau_{allowable}$) =68.75 MPa

Area (A) = 2 * (π/4) * d^2 (9)

= 2 * ((π/4) * (50^2)

A= 3926.99 mm^2

τ_{ind} = F/A (10)

= 76335.19/3926.99

τ_{ind} = 19.438 MPa < $\tau_{allowable}$

ii. Checking for Bending Failure

Allowable Bending Stress ($\sigma_{b\,allowable}$) = 137.5 MPa

UDL on pin (w) = F/l_e

= 76335.19 / 220

w = 346.978 N/mm

Maximum Bending Moment (M),

= [w * $(l_e)^2$] / 8

= [346.9*(220^2)]/8

M = 2099216.9 Nmm

Moment Of Inertia(I) = [π/64] * d^4

= [π/64] * $(50)^4$

I = 306796.15 mm^4

Maximum Bending Stress,

(σ_{bmax}) = [M * y] / I

= [2099216.9 *25]/ 306796.15

σ_{bmax} = 177.01 N/mm^2

σ_{bmax} < $\sigma_{b\,allowable}$

B. Arm :

Material- IS2062 ($\sigma_{t\,allowable}$ =102.5 N/mm^2)

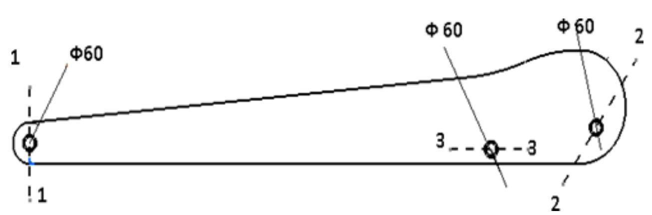


Fig. 6 Design of Arm

Section 2-2

Load	(Q) = 39839.83 N
Diameter of Hole	(d) = 60 mm
Width subjected to failure	(B) = 3.5*d= 210 mm
Thickness	(t) =20 mm

Checking for tensile failure

σ_t = P/A

= [Q/2] / [(B-d)*t]

= [19919.92] / [(210-60)*20]

σ_t = 6.64 N/mm^2 < $\sigma_{t\,allowable}$

C. BOOM :

Material- IS2062($\sigma_{t\,allowable}$=102.5 N/mm^2)

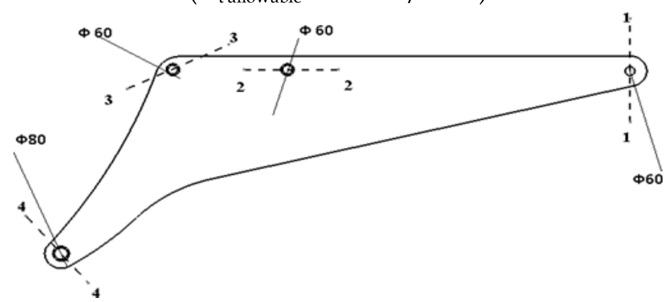


Fig. 7 Design of Boom Arm

Section 3-3

Load	(Q) = 40324.53 N
Diameter of Hole	(d) = 60 mm
Width subjected to failure	(B) = 2.5*d=150 mm
Thickness	(t) =20 mm

Checking for tensile failure

σ_t = P/A

= [Q/2] / [(B-d)*t]

= [20162.26] / [(150-60)*20]

σ_t = 11.2 N/mm^2 < $\sigma_{t\,allowable}$

D. BASE

Material- IS2062($\sigma_{t\,allowable}$=102.5 N/mm^2)

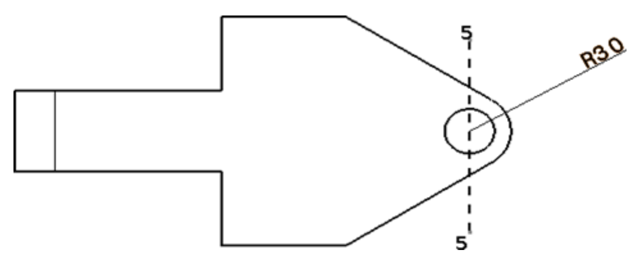


Fig. 8 Design of Base

Section 5-5

Load	(Q) = 86150.695 N
Width subjected to failure	(B) = 80 mm
Diameter of Hole	(d) = 50 mm
Thickness	(t) = 25 mm

Checking for tensile failure

σ_t = P/A

= [Q/2] / [(B-d)*t]

= [43075.3] / [(80-50)*25]

σ_t = 57.43 N/mm^2 < $\sigma_{t\,allowable}$

5. RESULTS:

CAD Model Design 1 :

TABLE No. 1 Stress Analysis of Various Pins

Sr. No	Part Name	Mathematical Calculated Result		Analytical Result	
		τ (ind) N/mm²	σ (b ind) w.r.t le (N/mm²)	τ (ind) N/mm²	σ (b ind) w.r.t Total length (N/mm²)
1	Pin 1	10.46	72.52	15.61	99.68
2	Pin 2	1.22	3.25	1.376	10.51
3	Pin 3	5.89	11.79	3.31	26.03
4	Pin 4	2.44	3.24	2.2	24.48
5	Pin 5	6.05	16.15	2.06	21.09
6	Pin 6	12.1	32.27	7.62	49.62
7	Pin 7	8.79	46.89	12.99	102.18
8	Pin 8	11.79	33.68	10.69	58.09
9	Pin 9	14.7	58.81	18.07	112.42
10	Pin 10	14.73	98.22	17.62	118.78
11	Pin 11	22.04	125.95	1.923	23.54

TABLE No.2 Stress Analysis of Various Parts

Sr. No.	Part Name	Section	Calculated σ_t (N/mm²)	Analytical σ_b (N/mm²)
1	Arm	1-1	0.956	1.32
		2-2	4.76	4.01
		3-3	1.912	1.42
2	Boom	1-1	1.49	2.05
		2-2	4.89	4.02
		3-3	9.78	1.42
		4-4	5.4	3.98
3	Base	1-1	10.35	7.005
		2-2	18.906	7.78
		3-3	9.424	9.34
		4-4	20.78	16.09

CAD Model Design 2 :

TABLE No. 3 Stress Analysis of Various Pins

Sr. No	Part Name	Mathematical Calculated Result		Analytical Result	
		τ (ind) N/mm²	σ (b ind) w.r.t le (N/mm²)	τ (ind) N/mm²	σ (b ind) w.r.t Total length (N/mm²)
1	Pin 1	10.463	79.521	15.61	99.68
2	Pin 2	1.80	10.8021	2.71	17.95
3	Pin 3	3.051	18.309	3.69	26.172
4	Pin 4	10.268	28.75	11.87	45.75
5	Pin 5	7.424	20.788	11.17	23.55
6	Pin 6	13.275	37.171	12.55	48.145
7	Pin 7	6.787	21.331	14.45	39.23
8	Pin 8	10.547	21.095	23.99	59.46
9	Pin 9	19.43	139.945	36.42	162.23
10	Pin 10	19.438	171.060	62.86	175.41
11	Pin 11	11.25	67.58	10.55	23.87

TABLE No.4 Stress Analysis of Various Parts

Sr. No.	Part Name	Section	Calculated σ_t (N/mm²)	Analytical σ_b (N/mm²)
1	Arm	1-1	1.2624	1.508
		2-2	6.64	0.84
		3-3	1.189	2.78
2	Boom	1-1	2.15	1.72
		2-2	4.859	2.69
		3-3	11.201	3.69
		4-4	8.619	4.58
3	Base	1-1	11.87	3.89
		2-2	16.89	12.16
		3-3	18.894	9.42
		4-4	7.0685	12.26

6. CONCLUSIONS:

Finite element modeling analysis showed that, the maximum stress induced does not exceed the material yield and tensile strength. Also the pin for lower and upper base, material can be changed to achieve greater strength.

The rotation of Hydraulic arm is achieved as per the requirement of company. This design would not only save the cost but also the time of the industry. The following things mentioned below shows the requirement of the company which is achieved:

- By Analytical and calculated method the hydraulic arm design found to be safe.
- Some of the pins should be made up of Hardened Steel so that it can survive for longer period.
- The design total reach are achieved as per the requirement.
- The hydraulic arm rotate about 60 degree for both of the designs.
- The Hydraulic arm can lift the scrap up to 200kg.

7. ACKNOWLEDGMENT:

We would like to thanks Prof. Sagar A. Dhotare for providing constant support throughout the work.

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