DESIGN AND ANALYSIS OF LOADER ATTACHMENT OF 13 TONNE EXCAVATOR FOR TUNNELING

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ABSTRACT

Excavators are heavy construction equipment consisting of a boom, arm, bucket and cabin on a rotating platform. All movement and functions of a hydraulic excavator are accomplished through the use of hydraulic fluid, with Hydraulic cylinders and Hydraulic motors. The excavator movement is actuated by an undercarriage with huge rubber tires or by means of a crawler track. In today's ever growing pace, construction operations, such as digging, ground leveling, carrying loads, dumping loads and straight traction play a very significant role and hydraulic excavators are needed to reduce manpower and complete the job in stipulated time more efficiently. Excavators provide solutions for handling materials within the confined areas of a tunnel. Purpose of tunneling may be for pedestrian walking, motor driving, rail travelling or even for making a path for water or oil transport through canal. In addition to this few tunnels are used for routing optic cables for signal transmission or cables for transmitting power or in some case used as sewers to transmit waste. The effective use of expensive construction machines has made it necessary to improve the performance characteristics of machines such as operating reach parameters and fault free operation within a tunnel. All these require for the study of the kinematics of the specified construction equipment. The project is carried out with respect to an existing 13 Ton class excavator manufactured by L&T - KOMATSU Limited, Bangalore. The kinematics of the excavator needs to be altered suitably so that it can work within the tunnel of 5 meter dimension (customer requirement). The aim of the project is to study the kinematic reach and forces developed, create a conceptual design, modeling using CATIA and Pro/Engineer and the analysis of the design using ANSYS.

KEYWORDS: Loader Excavator, Tunneling, Vector Loop Method, Finite Element Analysis

Received: Jun 09, 2020; **Accepted:** Jun 29, 2020; **Published:** Aug 06, 2020; **Paper Id.:** IJMPERDJUN2020649

INTRODUCTION

Iron ore mining, coal mining, excavations, constructions of roads, tunneling etc. are all bits and pieces which play a prominent role in developing nations for economic growth. Improving infrastructures are of prominent value for any country. Usually hydraulic excavators are preferred for this purpose. The hydraulic excavators are the machines, which are powered by the hydraulic power for the earth excavation. The major components of a hydraulic excavator are the Boom, Arm, Bucket, the Hydraulic cylinders that deliver the forces for excavation purpose. These machines are well known for their mobility and are excellent machines for general-purpose work, such as digging trenches and pits. Due to hydraulic power these machines exert huge positive force that helps the bucket to penetrate the material to be excavated.

A tunneling operation involves only material handling process, a loader excavator is chosen as a loader can exert much higher thronging force and a higher capacity bucket can be used when compared to the class of back hoe excavator. The major advantage of the shovel to form its own path as it moves forwards makes it the most suitable

excavator when digging above track level.

Figure 1: General View of Excavator.

Figure 2: Selection of Excavator.

KINEMATIC REACH AND FORCE ANALYSIS OF AN EXCAVATOR

The excavator consists of three different independently controllable mechanisms, making it is a three-degree of freedom system. Boom cylinders are the actuators of first mechanism that is rotation of boom. The actuator thus forms an inverted slider-crank mechanism which is relative to the frame. The second mechanism is also an inverted slider crank which is used to rotate arm or dipper stick relative to boom. The last mechanism is for bucket rotation, as these buckets are large a combination of four-bar and inverted slider-crank mechanisms in series are used, to cause outward and backward motions during the excavation process.

The vector-loop method is a classical procedure that can give a set of solvable vector equations, which can be solved either graphically for the kinematics of a planar mechanism. Vector loops around the linkage can be created by a position vector in a mechanism. In a loop, the vectors in that loop take us from one link through a joint, to another link, and another joint, and so on until we return to the same link that we started from. Vector loops lead to algebraic equations that can be solved for the kinematics of a mechanism. In a sliding joint (the hydraulic cylinders) in a mechanism, we must define a variable-length vector along or parallel to the axis of the joint. The link configuration, proper length and excavator loader working space are adequately obtained by position analysis.

Figure 3: Free Body Diagram of Excavator.

The formulae mainly used to solve the vector loops are the distance formula and the cosine rule for finding out the angles. The pivot points of each point on the mechanism are known and distance formulae is applied as

$$R_1 = \sqrt{(a_x^2 - b_x)^2 + (a_y^2 - b_y)^2} \text{ and } R_2 = \sqrt{(c_x^2 + c_y)^2}$$

There are five vector loops that are formed. Each of the vector loops is solved and the resultant magnitude and angles are known. These vector loops leads to the formulation of the bucket tip co-ordinates given by

The values of these were incorporated in Microsoft Excel in order to deduce the various Reach Parameters of the excavator.

Input Parameters (co-ordinates)				X	Y
1	Boom Length	3000	Cylinder foot point from swing center (X,Y,WG)	-585	1008
2	Arm Length	2300	Boom foot point from swing center (X,Y,WG)	-150	1410
3					
4					
5					
6	Vectors	Magnitude	Angle	Distance b/w Boom cyl pvt point to Boom foot pin (X,Y,WG)	2025 120.745
7	R 1	592.3081908	42.742196		
8	R 2	2028.596543	84.829541	Distance b/w Arm cyl pvt point to Boom foot pin (X,Y,WG)	1081.72 -184.53
9	R 3 (Boom Cylinder)	2500	253.6758		
10	R 4	3000	88.221894	Distance b/w Arm cyl pvt point to Arm foot pin (X,Y,WG)	626.382 -150
11	R 5	644.0817078	145.82779		
12	R 6 (Arm Cylinder)	2300	280.23074	Distance b/w Bucket cyl pvt point to Arm foot pin (X,Y,WG)	851.382 -150
13	R 7	1097.345531	271.65423		
14	R 8	1900.009474	132.17981	Distance b/w Bucket link pvt point to Arm foot pin (X,Y,WG)	1900 6
15	R 9	864.494872	122.15241		
16	R 10 (Bucket Cylinder)	1370.664924	318.4278	Boom Cylinder Range (°)	1750 2500
17	R 11 (Link A)	450	178.5308	Arm Cylinder Range (°)	1325 2300
18	R 12 (Link B)	475	91.583211	Bucket Cylinder Range (°)	875 1450
19	R 13	271.737	-44.335768		
20	R 14	400.0449975	312.17981	Link A Length	450
21	R 15	1541	90.000451	Link B Length	475
22	R 16	1385	281.81437		
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Figure 4: Pivot Points Calculation using Microsoft Excel.

Figure 5: Graphical Co-Relation.

To calculate reaction forces at various locations in an excavator during excavating process, a custom made program of L&T-Komatsu that is GW basic program is used. The maximum bucket tangential force is the crowding force generated by the bucket cylinder and tangent to the arc of radius.

Table 1: Reach Parameters

Product (Boom\ Arm Parameters Size)	13Tonne Loader (Tunneling) (3.0m\2.3m)
Maximum Reach (A)	6.6m
Maximum Cutting Height (B)	7.6m
Maximum Digging Depth (C)	2.07m
Maximum Dump Height (D)	4.36m
Max Reach at Ground (E)	6.28m
Reach at Cutting Height (F)	1.6m
Level Cut	3.16m
Minimum Swing Radius	1.98m

Figure 6: Envelope Diagram.

Figure 7: Force on Bucket Tip.

The crowding force obtained is 8659.18 kg (84.9kN at 111.81° with respect to front cutting edge of bucket tip.

Figure 8: Forces Acting on Different points of Boom.

Figure 9: Forces Acting on Different Points of Arm.

Figure 10: Forces at Different Locations on Arm.

Figure 11: Forces at Different Locations on Arm.

The Forces acting on each Point of the Boom and Arm are resolved into horizontal and vertical components. Using these forces the Axial force, Shear force, Bending moment, Minimum Sectional Area required and Minimum Section Modulus required at each locations are calculated. These values are used in designing of the Boom and Arm.

Table 2: AF, SF, BM, SA, SM for Boom

Position	Axial Forces N	Shear Forces N	Bending Moment N-mm	Sectional Area Required mm ²	Section Modulus Required
Arm Boom Point	342286.6	44880.5	0	1836.40	0
Boom Cylinder Point	468957.4	-2202.7	-2.85X10 ⁷	2516.00	1.53 X10 ³
Arm Cylinder Point	68988.1	-109449	3.50 X10 ⁷	370.12	1.88 X10 ³
Turret Point	-68992.0	109430	0	370.14	0

Table 3: AF, SF, BM, SA, SM for Arm

Position	Axial Forces N	Shear Forces N	Bending Moment N-mm	Sectional Area Required mm ²	Section Modulus Required
Arm-Bucket Point	425929.01	-77097.28	0	2285.14	0
Linkage Point	390237.13	171509.20	3.25 X 10 ⁷	2093.65	1.74 X 10 ³
Bucket Cylinder Point	-61145.58	58966.85	-2.98 X10 ⁸	328.05	1.60 X 10 ⁶
Arm Cylinder Point	37700.83	343160.71	-2.54 X10 ⁸	202.26	1.36 X 10 ⁶
Boom-Arm Point	-37702.42	343151.46	0	202.27	0

DESIGN

A detailed study was made on the previous models of the excavators to know the structural design of Boom and Arm for the best utilization of the existing components. Based on the conclusion from the detailed study the following designs of Boom and Arm are suggested.

ANALYSIS

The continuity of the meshing is maintained even at the faces where the cross-sectional areas vary. This property of continuity has to be maintained throughout the volume and is an essential feature for accurate results. The pink colored area in the figure depicts the places where the coupling equations are used.

CLOSURE

This paper presents the Design and Analysis of Loader Excavator of 13 Tonne class. Kinematic reach analysis was done by Vector Loop Method. Microsoft Excel was used to optimize the Pivot Locations. A Boom length of 3.0m and Arm length of 2.3m was found to be optimal for a tunnel size of 5m. Reach parameters of excavator for new attachments is found by using Microsoft Excel. Structure details were worked out through GW basic program, pressure in the cylinders and forces at various locations were estimated. The Boom was modeled using Pro/E Wildfire 5.0 and the Arm was modeled using CATIA V5 R18.

Ansys was used for analysis of stresses for critical loading condition. The maximum deflection encountered in the Boom was 1.844mm, while in the Arm it was 4.8mm, both of which were well within the safety limits of 6 to 7mm. The maximum Stress region in the Boom was found at the joint of the Arm Casting and Top Plate. The maximum value of which was 271N/mm^2 . In the Arm analysis, the maximum stress region was found at the joint of the Bucket Casting and Top Plate, at a value of 261N/mm^2 .

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