Design and Analysis of 22 Ton Class Excavator Swing Braking System

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Abstract:- The present work is carried out to avoid the field failures and to get better service life of swing machinery components and upper structure. The main objective is to find the inertia force acting on the system. This work helps in strengthening and selection of the swing machinery brake system and the upper structure so that field failure will be minimized, downtime of the equipment is reduced and to increase the productivity, availability, saleability of the excavator. The tool used for designing the component is Inventor for Modelling software and analyse the effect of force by Ansys workbench software. Later ADAMS software is used to study the kinematic analysis and dynamic analysis of the excavator moving parts. The result analysis of existing design indicates improvement in design efficient for the application of 22 Ton excavator swing braking system.

Keywords:- *Hydraulic motors, secondary control, position control, speed control ,excavator, swing.*

I. INTRODUCTION

An excavator are heavy vehicle used in earth working, lifting of heavy parts, demolition and handling of materials. It consists of body of the excavator where operator works with wheels under it, track and attachments above the body part consists of arm, boom and bucket to work. Attachments with the arm depends on purpose of usage. The movement of rotation is called swing where hydraulic driving system is used. The excavator is normally a diesel engines where many hydraulic pumps are driven. This pumps supplies oil flow to the system that operates the different working function along with driven transmission. This swing is hence driven by the hydraulic motor that creates motion of rotation that can rotate upto 360 degrees. When it is used for earth working its characteristics is to dig at one position and rotate for required angle, dump the load and move back.
A heavier load will be noticed by the operator since pressure needs to built up for a long time hence the joysticks operation will feel different. This is the reason for open centre systems are still employed in larger excavators. Another feature in separate circuit for the slew drive consisting of a closed loop hydrostatic transmission(HST). The swing can be controlled without values and with less throttling losses. Separation of slew gear from the working hydraulics prevents mutual influences in case of parallel movements. This type of system is also common in heavier excavators where the loads are high.

Modern hydraulic excavators comes in a wide variety of sizes. The smaller ones are compact excavators. This weighs around 930kgs. The largest model CAT 6090 weighs around 979,900 kgs, and bucket as large as 52.0 m^3. Weights of the attachments will depend upon the purpose of use. For different purpose different types of excavators are used.

II. LITERATURE REVIEW

Zhang Wobo and et.al., constituted the solid model of SLWY-60 excavator and the work and motion state model of excavator was build based on unigraphics(UG) and transmitted into Automated Dynamic Analysis of Mechanical System(ADAMS) [1]. The size and positions of work devices were parametric. Stimulation model for work state was established by means of defining different joints and motion driving.

G.Wszolek presented a numerical vibration of analysis of the excavator model. This model with a discrete distribution of parameters was attracted to kinematic and dynamic exhibitions [2]. The analysis was made in GRAFSIM program.

A flexible multibody system power of the excavator process of the hydraulic excavator was presented by Imanishi and et.al., in their research [3]. It was obtained that excavator working process of the deformation characteristics of the device and is verified the dynamics simulation for the excavator design and performance analysis for the whole machine.

Zygmunt Towarek discussed that the dynamics of a spatial model of a single bucket excavator on a caterpillar chasis [4].The strain of foundation is being taken into consideration.

Research work carried out by Shi Qinglu and et.al., conducted comparative analysis on the intensity of the excavator boom by finite element method [5]. The analysis results shown that local stresses of the boom were relatively higher under offset loads and transverse loads. There are higher stress at some parts working under two conditions.

Naresh N.Oza carried out the finite element analysis (FEA) and optimization of earth moving attachment as backhoe.[6]

Yang Xianping as studied on Hydraulic excavator structure model in virtue of Ansys [7]. The nature frequency and the model of vibration were studied to access its dynamic characteristics by model analysis results and found that the connecting shafts at models was easily to produce abrupt, thereby enhancing the performance of hydraulic excavator.

III. PROBLEM DEFINATION

During the swing operation of excavator the machine should stop within 35 degree angle after leaving the joystick by the operator. But that is not happening due to the inertia force which is high. Hence stopping angle shift will be experienced which will affect the productivity of the machine.

IV. METHODOLOGY

To carry out the correct method for the analysis of excavator, large amount of data is required. These data's helps to understand the complete working principle of the excavator so that further development of the excavator can be done.

Have complete information of the excavator attachment parts from the manual provided by BEML of 22 Ton class excavator machine.

The softwares to be followed for the complete analysis of the excavator is shown below.

- INVENTOR for modelling the parts (3D)
- ANSYS Workbench for meshing the parts
- ADAMS for kinematic and dynamic analysis of the excavator machine.

Following steps shows the complete methodology of the present work.

A. Part Modelling

Part modelling is done through 'Inventor' software. From the survey, literature review, review from the experts, review from the design engineers of the excavator manufacturing industries, working range and the dimensions provided by BEML is used for modelling. Length of the attachments to the body part of the excavator is considered. Material selection for the arm, boom and bucket is provided by manufacturing industries as per their requirements.

➢ *Design of bucket*

Bucket Geometry: Geometry of the 22 Ton class

- excavator machine will be provided by the companies manual. The length, width, height of the bucket will be mentioned in the manual. Modelling is done using Inventor software with provided dimensions and part is renamed as bucket.
- Material Selection: Standard material selection can be done as per companies requirement. This material selection will be based on strength to weight ratio, characteristics of the parts, purpose, amount of load impact on the bucket. The material is selected in library within the software which provides various types of materials option to choose.
- *Bucket Design*: Bucket designed should withstand the loads and can operate smoothly in the required working condition. Surface finishing must be smooth at corners and edges. The design should operate at different positions and angles (Ex: digging and lifting) without any failure.

➢ *Arm and Boom design*

The arm and boom design process also involves the same method as mentioned above for bucket design. Material will be selected as per companies requirement.

➢ *Design of other parts*

Other parts of the like pins at boom, arm, bucket joint etc. will be designed based on strength and rigidity considerations using FEA approach.

➢ *Modelling of body part*

Body part is modelled based on companies standard dimension and market requirement. Material selection is based on requirements during kinematic and dynamic analysis. Weight distribution will be 40% of weight is added to the body where 60% of weights are added to the attachments.

B. Finite Element Analysis

Methodology adopted for meshing is as follows.

- Model has been imported from inventor software as an "parasolid" file.
- Selecting the plane.
- Geometrical errors: Small holes, Fillets, Chamfers and other such geometries which are not important for structural analysis are neglected.
- Components are meshed with default settings or with fine mesh using 2D and 3D elements.
- Specified quantity criteria are followed for meshing.
- Boundary conditions are applied using 1D elements.
- Material properties are assigned using library to the FEA model.
- FEA model has been solved.
- Results are interpreted.

  Following are the standard properties and values used during FEA analysis.

  Material property: Structural steel/ SALIMA 450HI

Sr. No.	Property	Value
1	Poisson's ratio	0.3
2	Modulus of elasticity	210000N/mm
3	Density (Rho)	7900 Kg/m³
4	Yield stress	450a

Table 1

C. Kinematic and Dynamic Analysis in Adams

The kinematic and dynamic analysis is done in ADAMS software. The parts modelled in Inventor should be assembled and save in Parasolid format. Saving the model other than parasolid file will make the complete model into single part which cannot be further analysed in ADAMS. Later the model is transported into Ansys workbench for meshing and then file is opened in ADAMS.

- *Importing to ADAMS*

The geometrical model should be in parasolid format which is imported to ADAMS, where a working environment is made similar to the real environment by setting the unit in MMKS system and gravity to -Y etc. Model is shown in the fig 1. After importing add constraints and driving function. Define constraints to STEP function.

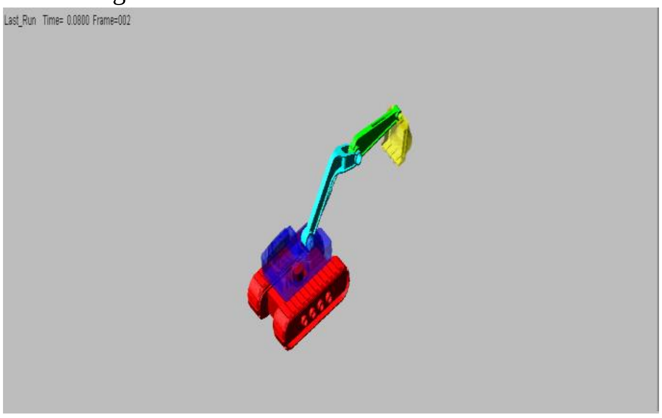

Fig 1:- Excavator model in ADAMS

- *Kinematic analysis*

First the body of the excavator is fixed to ground. Then the body is fixed with respect to the attachments of the excavator. The movement of the excavator attachments is programmed. Geometry position of backhoe device is determined by the hydraulic cylinder lift arm. Adjust bucket rod cylinder in full shrink state, where motion function: step(time,0,0,5,500). Adjust the bucket cylinder in full shrink state, motion function: step(time,5,0,10,550).

Movement function of hydraulic cylinder is step (time,10,0,20,-160)+step (time,20,0,40,160+750). Simulate results and draw curves. Trace marker of arm is shown in fig 2 .

Fig 2:- Trace marker of arm

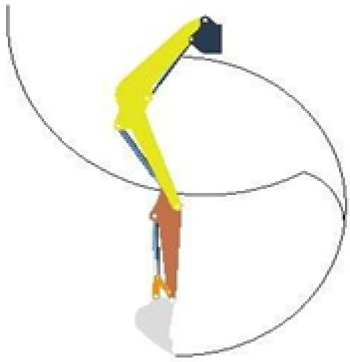

Fig 3:- Trace marker of bucket rod

Adjusting the boom cylinder in the full shrink state, motion function: step(time,0,0,15,750). Adjusting the connecting point between the boom and bucket rod, the connecting point between the bucket rod and bucket, motion function: step(time,15,0,20,420).

Movement function of bucket hydraulic cylinder is step(time,20,0,35,-415)+step(time,35,0,50,815).

Transfer ADAMS to replay simulation results and draw curves. Trace marker, trajectory chart of mark point is shown in fig 3.

- *Dynamic analysis*

Simulation is carried out to ensure the given constraints are correct to the excavator. Dynamic simulation helps to understand the load carrying capacity of the parts at different working environment and at various external loads. The result obtained is used to study for the further improvement. This study helps to avoid the field failures and to improve the service life of an excavator. Dynamic simulation is carried out to find centre of gravity, torque and inertia of the excavator machine.

V. RESULTS AND DISCUSSION

To evaluate the results, we need to compare the results with motions that skillful operator made. For this, we measured real motions of excavator which is made by skilful operator. The measurement data contains excavator lengths, forces, swing angle and swing torque.

A. *Results*

The results obtained in ADAMS are depicted in Fig 4. and Fig 5.

The graphs in Fig 4. represents the maximum reach and maximum lift point of excavator attachments during digging and lifting process.

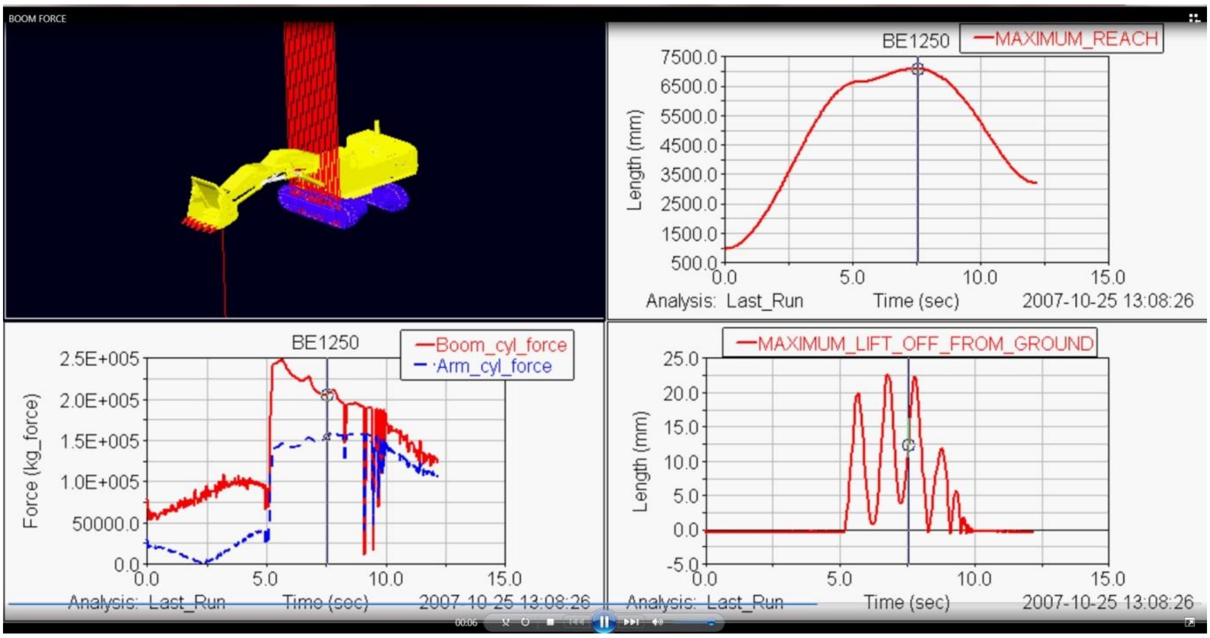

Fig 4:- Excavator boom and arm force

Fig 5. shows the boom and arm force of excavator machine with respect to time. The graph represents time(sec) in x axis and force (kg/force) in y axis. When the boom and arm is rotated at 90 degrees in digging process the force become maximum at seconds, resulting 2.5*10^5 force/kg. Hence this results in overshooting of the arm and boom in that point. Hence resulting poor exhibition of working.

This represents the centre of gravity (CG) of the CM marker at position x and at position z axis. Analysis is run for 18 seconds in ADAMS.

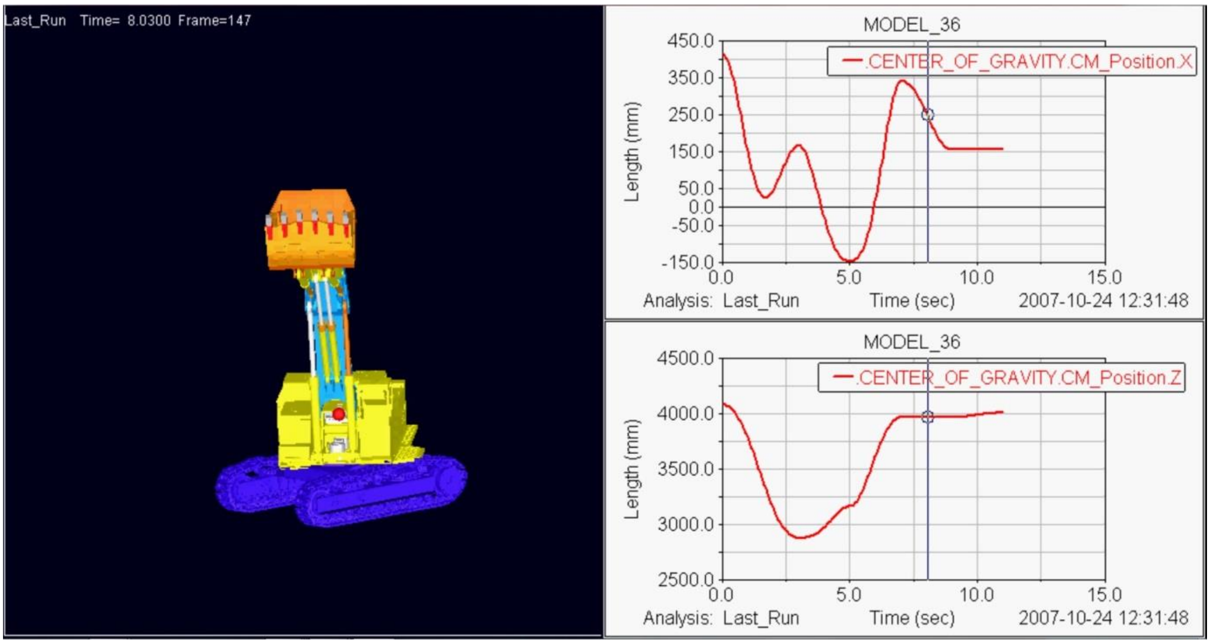

Fig 5:- Centre of gravity of excavator

To remove the overshooting of the boom and arm, the overall weight distribution of the excavator machine should be changed. This can be achieved by trial and error method of weights carrying individual excavator parts of excavator machine.

Final weight distribution of the excavator is found are listed in table.

PART	WEIGHT (KG)
ENGINE	655
DAMPER	12
MAIN PISTON PUMP	116
RADIATOR ASSEMBLY	130
HYDRAULIC TANK	196
FUEL TANK	111
REVOVLING FRAME	1864
OPERATORS CAB	395
SWING MACHINERY	230
SWING MOTOR	73
SPEED CONTROL VALVE	195
COUNTER WEIGHT	3935

Table- 2

Obtained swing torque value is 65KN. Hence machine availability can be achieved more than 85%. During weight distribution of excavator parts it is found that overall weight of the excavator attachment should be 60% to the counter weight, where the counter weight should be 40 to the overall body weight of the excavator machine.

B. *Calculations*

For various parameter calculations the following formula and explanation shown below.

- *Inverse dynamics*

Inverse dynamics are used to find joint torque from position, velocity and acceleration of each joint. Since solving problems with closed loop is not easy, cut the closed loop as seen in Fig. 6. and adapt open loop recursive dynamics algorithm.

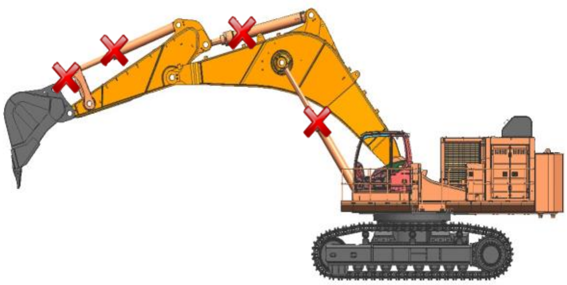

Fig 6:- Reduced system of excavator

Dynamic formulation can be stated as below

$$\hat{\tau}=\tau+J^T F=M(q)\dot{q}+C(q,\dot{q})$$

M represents the mass matrix constituted as momentum of inertia and mass. C implies centripetal and coriolis force. J^T *F* is the external force of excavator which is the constraint force for compensating virtual open chain. q is the configuration matrix., inverse dynamics is calculating for given cylinder motion.

If we let reduced system torque as □□ˆ, the relation of original system torque yields

$$\hat{\tau} = \tau + J^T \lambda$$

Constraint jacobian *J* satisfies the relation of

$$V = J\dot{q}$$

- *General friction law*

The coefficient of friction between two surfaces in contact is equal to the force required to overcome the friction divided by the reaction force between the two surfaces as shown in the

$$\mu = \frac{F}{R}$$

where, μ= Co efficient of friction
F= Force required to overcome the friction
R= Reaction force between two surfaces

Therefore, the retarding torque of shoe brakes is a product of the effect of the torque spring pressure and the coefficient of friction of the lining material.

- *Braking torque*

Brake torque is the force applied to the brake wheel to stop the motion of the moving equipment.Assuming the operating conditions for the equipment are constant, a brake having a retarding torque equal to the full load torque of the motor to which it is applied is usually satisfactory.

The torque can be determined from the following formula for both AC and DC motors.

$$T = \frac{\frac{5250 \times HP \times}{SF}}{RPM}$$

Where, T = Brake Torque (Lb. Ft.)
5250 =Constant
HP = Motor Horse Power
RPM = Speed of Brake wheel
SF = Application Service Factor

The brake torque selected might be greater or lesser that depends on the type of application.

VI. CONCLUSION

Based on methods and analysis carried out on 22 ton class excavator swing braking system the following conclusions are drawn.

Static and dynamic analysis is calculated using ADAMS after importing the geometrical shape of excavator in parasolid format from the Autodesk Inventor.

ADAMS used to design the break clutches, total number of clutches required, size of the clutches required during machine swings at 90 degrees and 360 degrees can be solved.

By considering weight of the hydraulic tank required, engine weight, attachment of bucket weight is calculated and resultant dynamic torque is arrived.

Swing torque value obtained is 65KN. User will adapt due to lot of warranty claims. Failures will be reduced. User will appreciate if the machine availability is more than 85% throughout this year. Hence machine availability should be achieved more than 85%.

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