National Conference "CONVERGENCE 2016", 06th-07th April 2016

Review of Finite Element Analysis of Crane Hook

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Abstract

A Crane hook is a device for grabbing and lifting loads of heavy duty by mean of a device such as crane hoist. A crane hook is usually equipped with a safety latch to prevent the disengagement of the lifting rope to which the load is being attach. The CAD model of a lifting crane hook is initially prepared from the Analytical Design.

Keyword :- crane hook, FEA.

1. INTODUCTION

Crane hooks are the components which are generally used to elevate the heavy load in industries and constructional sites. Recently, excavators having a crane-hook are widely used in construction work sites. One reason is that such an excavator is convenient since they can perform the conventional digging tasks as well as the suspension works. Another reason is that there are work sites where the crane trucks for suspension work are not available because of the narrowness of the site. In general an excavator has superior maneuverability than a crane truck. However, there are cases that the crane-hooks are damaged during some kind of suspension works. From the view point of safety, such damage must be prevented. Identification of the reason of the damage is one of the key points toward the safety improvement. If a crack is developed in the crane hook, mainly at stress concentration areas, it can cause fracture of the hook and lead to serious accidents. In ductile fracture, the crack propagates continuously and is more easily detectable and hence preferred over brittle fracture. In brittle fracture, there is sudden propagation of the crack and the hook fails suddenly. This type of fracture is very dangerous as it is difficult to detect

Crane hooks are one of the important components used in industries to carry heavy loads basically crane hooks are designed with pulley elongated by rope or a chain. In this paper mathematical calculations are done by varying cross sections triangular, trapezoidal, and rectangular with constant area of cross sections for two different materials of steel, cast-iron analysis are evaluated. PRO/E design is prepared, for different cross sections by preparing the trajectory with the parameters taken from the design data book. In this project result of mathematical calculations is compared with the ANSYS with constant area of cross section, modifying the type of material optimization of better results were done inTaguchi Method.

2. LITERATURE SURVEY

Mr. A. Gopichand[1] studied in this paper the Crane hooks are liable components subjected to failure due to stress in accumulation of heavy loads. The design parameters for crane hook are area of cross section, material and radius of crane hook.

ANALYTICAL METHOD FOR STRESS CALCULATION

Curved beam flexure formula is used when the curvature of the member is pronounced as in cases of hooks and rings for different cross sections mathematical analysis of stress

$$\sigma = \frac{F}{A} + \frac{M \times y}{I}$$

Where

M=maximum bending moment

Y=Distance between centroidal axis to neutral axis

I=Moment of inertia for different cross sections

Taken from design data book. A For a Triangular cross section

In the present work optimization of design parameters is carried out using Taguchi method, total three parameters are considered with mixed levels and L16 orthogonal array is generated .The optimum combination of input parameters for minimum Vonmises stresses are determined. From the Taguchi analysis the optimum combination of input parameters for von misses stress were determined using SN ration plots, The optimum combination is TRIANGULAR cross section ,CASTIRON material and 200mmRADIUS .The first model simulation values are compared with analytical values and the error is **9.26%** which is acceptable.

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Tejas P. Jani[2] studied in this paper The design of crane hook contains such parameters like cross section of hook, material and radius of curvature. We have selected the cross section as a basic parameter to optimize the hook design which carries 8tons load. The design is generated in Creo2.0 and analyzed in ANSYS 15.0 workbench (FEM). By applying modified cross section in place of Standard Trapezoidal section of hook and considering Stress analysis, Weight optimization is carried out. The modified design is further compared with Trapezoidal, Triangular and Circular hook. They have optimized the standard design applying various cross sections and the optimized cross section has been selected in which the stress is increased by 6.79% which is acceptable and the weight is reduced by 18% nearly which can be considered as optimize results.

Chetan N. Benkar[2] studied in this Real time pattern of stress concentration in 3D model of crane hook is obtained. Finite Element Analyses have been performed on various models of crane hook having triangular, rectangular, circular and trapezoidal cross sections.

Table1. Stress Calculations

Circular Cross	D = d = 256.2
Section	πd^2
	$A = \frac{1}{4} = 51526.1 mm^{\circ}$
	d
	$R = c + \frac{1}{2} = 428$ h^2
()-1	- 4290.90
	$W = \frac{1}{200.09}$
	$\sigma = \frac{W}{m} + \frac{M}{m} \left[1 - \frac{K}{m} \right]$
	$A^{-}AR\left[-h^{2}(R-y)\right]$
	Considering $M \approx W \times R$ we can have
	$W[$ R^2y]
	$\sigma = \frac{1}{A} \left[2 - \frac{1}{h^2 (R - n)} \right]$
	$\sigma = -368.7$ for $y = \frac{a}{2}$ & for y
	2 , , , , , , , , , , , , , , , , , , ,
	$=-\frac{a}{a}$
	2
Rectangular	D = 256.2 mm
Section	$A = 51562.1 \ mm^2$
· · · · · · · · · · · · · · · · · · ·	B = 120.4 mm
	$R^2 (2R + D)$
, Ŭ,	$h^2 = 2.3 \frac{1}{D} \log \left(\frac{2R}{2R} - \frac{R}{D} \right) - R^2$
	$b^{2} - 5626$
	n = 5020
	$\sigma = 12.50$ for $y = \frac{\alpha}{2}$
	2
Triangular Cross	D = 256.2 mm
Thangular Cross	D = 250.2 mm
Section	$A = 51552 mm^{-1}$
$\triangle \uparrow \downarrow_{a}$	B = 402.5 mm
, <u>∏</u> ,	$h^{2} = \frac{R^{*}}{R} \times \frac{B}{R} \left[2 2R \log \left(\frac{R_{2}}{R} \right) - R \right]$
	$n^{n} = \overline{D} \wedge \overline{D} \left[2.5 R_2 \log \left(\overline{R_1} \right) - D \right]$
	$-R^{2}$
	Considering $M \approx W \times R$ we can have
	$\sigma = 17.97$
Trapazoidal cross	P = 300 mm
riapezoidar cross	$B_1 = 500 mm$
section	$B_2 = 102.4 \text{ mm}$
	$=\frac{K^{*}}{2} \left[2 \cdot 3 \left(B_{1} - B_{2} \right) K_{2} \right) \log \left(\frac{K_{2}}{2} \right) \right]$
/ P \ /	$A \begin{bmatrix} a & b \\ a \end{bmatrix} = D = \int B B (R_1)$
\\'_	
	$ -(B_1 - B_2) - R^2$
91. R.	Considering $M \simeq W \times R$ we can have
	~ 16.25
	$\sigma = 10.35$

A] FAILURE OF CRANE HOOKS

Strain aging embrittlement due to continuous loading and unloading changes the microstructure. Bending stresses combined with tensile stresses, weakening of hook due to wear, plastic deformation due to overloading, and excessive thermal stresses are some of the other reasons for failure. Hence continuous use of crane hooks may increase the magnitude of these stresses and eventually result in failure of the hook. All the above mentioned failures may be prevented if the stress concentration areas are well predicted and some design modification to reduce the stresses in these areas.

B] THEORETICAL ANALYSIS

Machine members and structures subjected to bending are not always straight as in the case of crane

National Conference "CONVERGENCE 2016", 06th-07th April 2016

hooks, chain links etc., before a bending moment is applied to them. For initially straight beams the simple bending formula is applicable and the neutral axis coincides with the centroidal axis. A simple flexural formula may be used for curved beams for which the radius of curvature is more than five times the beam depth. For deeply curved beams, the neutral and centroidal axes are no longer coinciding and the simple bending formula is not applicable.

Govind Narayan Sahu[3] studied in this paper It is then followed by implementation of modified cross section of hook in the static structural analysis workbench of catia v5. These results lead us to the determination of stress and deflections in the existing model. In order to reach the most optimum dimensions several models in the form of different dimensions of hook were tested and the most optimum dimension was selected. The selection was based on the satisfaction of several factors in the form of load carrying capacity, stress induced and deflection. Stress induced and displacement in "Modified Curved Hook" is least because of curved shape and fillet edges as well as stress concentration are distributed uniformly. It have less mass due to this we are able to save the material and balance economy.

Rashmi Uddanwadiker[4] Did The complete study is an initiative to establish a FEA procedure, by validating the results, for the measurement of stresses. For reducing the failures of hooks the estimation of stress

The location of the neutral axis is obtained by equating the product of the normal stress and the area elements over the whole area to 0.



Fig 1 Curved beam with its cross section area.

$$\int_{A} \sigma dA = E \frac{d\phi}{\phi} \int_{A} \frac{r - r_{n}}{r} dA = 0$$

$$A = r_{n} \int_{A} \frac{dA}{r}$$

$$r_{c} = \frac{1}{A} \int_{A} r dA$$

$$M = E \frac{d\phi}{\phi} Ae \qquad \text{(where } e = r_{c} - r_{n}\text{)}$$

$$M = \frac{r \sigma Ae}{r - r_{n}}$$

$$\sigma = \frac{M(r - r_{n})}{\sigma Ae} = \frac{-M.y}{Ae(r_{n} - y)} \qquad \text{(where } y = r_{n} - r\text{)}$$

$$\sigma_{i} = \frac{M.c_{i}}{A.e.r_{i}}$$

$$\sigma_{o} = \frac{M.c_{o}}{A.e.r_{o}}$$

$$R = Ct(\sigma_{11} - \sigma_{22})$$

$$\varepsilon = (r - r_{n})\frac{d\phi}{r\phi}$$

The strain is clearly zero at the neutral axis and is maximum at the outer radius of the beam. Using the rela-tionship of stress/strain = E, the normal stress is simply.

$$\sigma = E\varepsilon = E(r - r_n)\frac{d\phi}{r\phi}$$

CONCLUSION

From the overall literature review it get concluded that in this way design the crane hooks and analyse that crank hook .from this relation we can design the crank hook to different loading condition and as well as analyse that hook under same condition

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National Conference "CONVERGENCE 2016", 06th-07th April 2016

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