Design and Analysis of Central Drum in Mine Hoist

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Abstract – *This paper presents design and analysis of central drum in Mine Hoist. Central drums in mine hoists are too large and heavy; because of this they are difficult for manufacture, installation and operating. Hence there is needed to make it light in weight without affecting its strength. The design optimization can be achieved by replacing the disc type structure at the side of central drum by arm type supporting structure. For this we designed different conceptual models and analyzed them under maximum loading condition. After comparing the results best design is selected and it will be found easily possible for manufacturing and operating.*

Key Words: Mine hoist, Central drum, Side disc, Arm type structure, easy for manufacture etc.

1. INTRODUCTION TO MINE HOISTS

In mining industry mine hoist is the most commonly used mining equipment used for hauling and conveying the mining material. These are commonly used in coal and iron mines etc. Mine hoists are mechanical equipment by using which mining material will be raised or lowered. Mine hoists works on principle that load can be lifted by applying equal and opposite force.

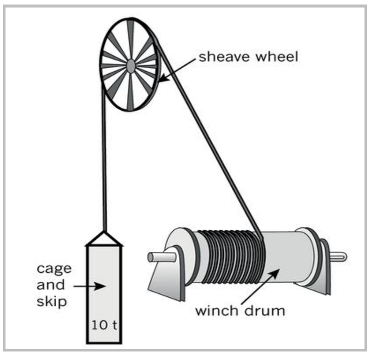

Fig – 1: Mine hoist

Mines are generally powered by using the electric motor. Motorized mine hoists have made mining more rapid. The most common type of mine hoist used in mining industry is drum type mine hoist. These are mainly consisting of central drum, sheave pulley and cage or skip. In drum type mine hoist wire rope is wound on central drum while the lifting the cage.

2. PROBLEM IDENTIFICATION

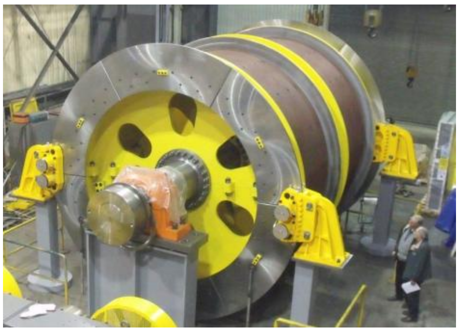

Fig – 2: Central Drum

Central drum in mine hoists is too large in diameter, hence it is difficult to manufacture. Also it is too heavy in weight; because of this it is difficult for manufacturing, installation and for operating with loads. Hence it is needed to make it possible to manufacture and working. Hence design will be made to make it little light weight with have no effect on its strength.

3. LITERATURE REVIEW

WANG Jiu-feng, XU Gui-yun, ZHU Jia-zhou, YANG Yan-chu, in their paper named as, "*Parametric Design and Finite Element Analysis of Main Shaft of Hoister Based on Pro/E*", advanced parametric design method which realized in the process of modelling of main shaft of hoister was deal by using the interface technology between Pro/E and ANSYS software. The simulation analysis of stress status of the main shaft of hoister designed in Pro/E under a certain load is made. The adoption of this method will dramatically shorten the development cycle and cut down the design costs. Otherwise the research method will reference value to gear model library development and to the optimization design of the main shaft of hoister. [1]

LUO Jiman, XING Yan, LIU Dajiang and YUAN Ye, in their paper, "*Modal Analysis of Mast of Builder's Hoist Based on ANSYS*", For the purpose of researching the factors which affect the dynamic characteristic of mast of builder's hoist and analyzing the impact of different factors over system security, the authors of the paper applied the finite element method to build the model and made the modal analysis for mast which was installed with various installation distances or under different working conditions. [2]

Yang Yuanfan, in the paper named as, "*The Study on Mechanical Reliability Design Method and Its Application*", Through the study on mechanical reliability design and combination with the structure of mine hoist, it is proposed that the crucial procedure of reliability design's application into mine hoist is as to ascertain the statistics of the relevant parameters, then to set up the failure mathematical model, and finally the reliability design can be operated. [3]

J.J. Taljaard and J.D. Stephenson, in the paper named as, "*State-of-art shaft system as applied to Palaborwa underground mining project*", The design of a 30,000 ton per day underground mine at Phalaborwa presented many and various challenges to the owner and the design team. Using modern best and proven practice, innovative engineering, extensive test work and verification by worldwide experts these challenges were met head on and overcome. The state-of-the-art system will be in operation by the end of the year 2000. [4]

Shuang Chen and Shen Guo, in their paper named as, "*Stress Analysis of the Mine Hoist Spindle Based on ANSYS*", In this paper, the three dimensional modeling of 2JK mine hoist spindle was established by using Pro/ E according to given data. Then the model was inputted into the finite element analysis in ANSYS, the stress distribution of the spindle was obtained, strength check of the dangerous section was made at the same time, which provides a accurate and reliable theoretical basis for improving the spindle structural design. [5]

HuYong and HuJiQuan, in their paper named as, "*Mechanical Analysis and Experimental Research of Parallel Grooved Drum Multi-layer Winding System*", In the present design criterion of multi-layer winding drum, multi-layer winding coefficient is chosen according to the number of wire rope layers. However, the actual wire rope arrangement on the drum and the elastic property of wire rope also play decisive roles in determining the multi-layer winding coefficient value. Analyzing the actual stress of the drum accurately is the precondition of ensuring the drums safety and reliability for meeting the lightweight design requirements. [6]

4. DESIGN AND CAD MODELLING

Our main objective of project work is to make central drum light in weight. For this we have designed different conceptual models replacing the disc shape side supporting structure of central drum with casting arms patterning circularly. This will make central drum little in light weight without affecting its strength. The actual and conceptual model of central drum will be as shown in fig – 3.

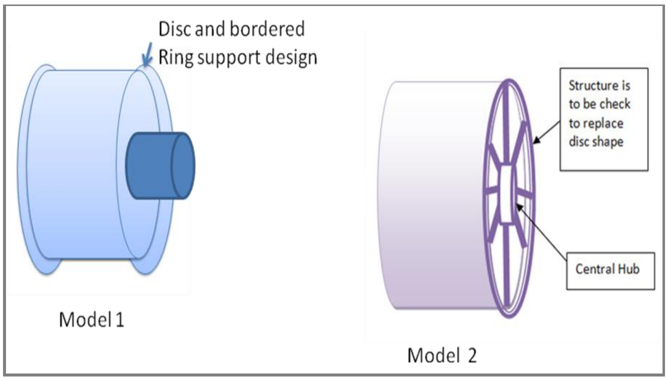

Fig – 3: Actual and conceptual model

4.1 DESIGN PARAMETERS

According to ASME code the maximum permissible shear stress (τ) will be,

$\tau = 0.3\ \sigma_{el}$ or $0.18\ \sigma_{ut}$

$\tau = 0.3\ \sigma_{el} = 61.5$ MPa OR $\tau = 0.18\ \sigma_{ut} = 93.6$ MPa

Where,

$\sigma_{el} = 205$ Mpa

$\sigma_{ut} = 520$ Mpa

Hence maximum permissible shear stress is $\sigma_{el} = 61.5$ MPa

The shaft will be subjected to twisting moment or torsion which will be obtained by using torsion equation,

$$\frac{T}{j} = \frac{\tau}{R} \quad \text{....... (i)}$$

Where,

T = torque acting on the shaft

j = polar moment of inertia

τ = torsional shear stress

R = Distance from neutral axis to outermost fibre

= D/2.... Where D is diameter of the shaft

We know that, polar moment inertia (j) is given by,

$$j = \frac{\pi}{32} D^4 \quad \text{........ For solid circular shaft}$$

Where,

Combined shock factor (K_t) and fatigue factor (K_m) are taken as 1, for the gradually applied or steady load.

Torque to be transmitted is T = 700 Nm

Hence, from eq. (i)

$$\frac{700\ x\ 10^3}{\frac{\pi}{32} D^4} = \frac{61.5}{D/2}$$

D = 38.7 mm ~ 40 mm

Also from torsion rigidity equation we have,

$$\theta = \frac{584TL}{GD^4} \quad .... \text{(ii)}$$

Where,
θ = angle of twist in degree
T = Torque, N-mm
L = length of shaft, mm
G = Modulus of rigidity, $N/mm^2 = 70.3\ kN/mm^2$
D = Diameter of shaft, mm
*Let the angle of twist for the shaft 1degree i.e. $\theta = 1^0$.

Hence,

$$1 = \frac{584 \times 700 \times 10^3 \times L}{70.3 \times 10^3 \times 40^4}$$

L = 440 mm ~ **500mm**

Rope specification
Length of rope: 50000 mm
Nominal breaking load: 133 kN
Weight: 0.86 kg/m
Rope construction: 6 x 26 RRL(right regular lay) rope
Safety factor of rope
= (Minimum breaking load) / Load applied
= 133 / 15 = 8.87

Drum calculations
Parameter required,
Maximum load =15 kN
Diameter of rope = 14 mm
Length of rope = 50000mm

Calculation

a) **Diameter of drum**
Ddrum = (ratio between 20 to 25) x drope
=20 x 14 = 280 mm
b) **Groove radius,**
r = 0.53 x d = 0.53 x 14mm = 7.4mm
c) **Groove diameter,**
d = groove radius x 2 = 7.4mm x 2 = 14.8mm
d) **Pitch diameter,**
p = 2.065 x groove radius = 2.065 x 7.4mm
= 15.281mm
e) **Groove depth**
h = 0.374 x d = 0.374 x 14.8 = 5.5352mm
f) **Thickness**
tx = P/kp = 15000/(210000000 x 0.0148)
= 4.826mm
g) Y= 520 mm
h) L5 = L6 = 20 mm
i) Number of layer = 3
j) Number of groove = 20
k) Drum grooved length,L3
L3 = (n - 1)P = (20 – 1) 15.281 = 290.24 mm
l) Drum un-grooved length,L1=L2
L1 = L2 = 1/2 diameter of hook + radius of rope
= 47.13 mm
m) Safe Factor = 6

4.2 3D MODELLING

We have done modelling of our project prototype in the Pro-E software which was user friendly to us. 3d model of project prototype is shown below.

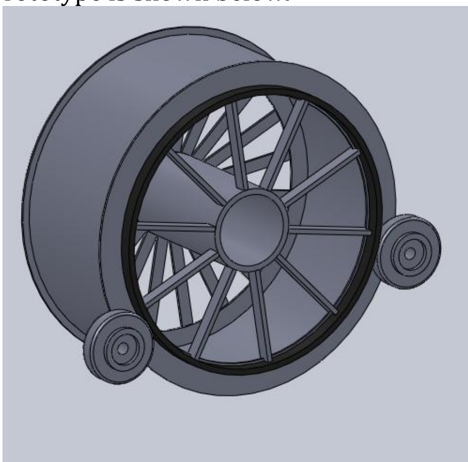

Fig – 4: CAD Model

5. ANALYSIS

The structural analysis of central drum is done by using ANSYS software. The ANSYS is a general purpose Finite Element Modelling package; it is used for numerically solving a variety of mechanical problems which include static and dynamic structural analysis (both linear and non-linear), steady state and transient heat transfer problems, mode-frequency and buckling analyses, acoustic and electromagnetic problems and various types of field and coupled-field applications.

First Step in Finite Element Method (FEM) involves Sub-division of a body or structure into Finite Elements known as descritization or meshing. Meshing of our project prototype is done by using 20 noded solid 95 elements. The meshed model of our project prototype is as below,

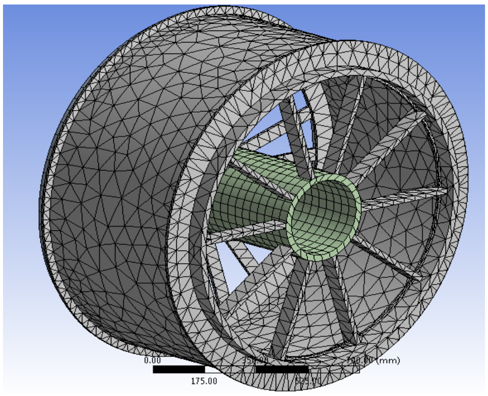

Fig – 5: Meshed model of central drum

The results are obtained by applying boundary conditions as follows,

(i) RPM = 10 Max.
(ii) Force = 5000N

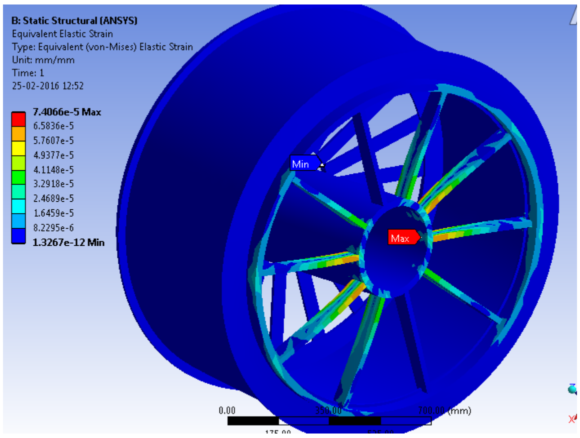

Fig – 6: - Equivalent Elastic Strain

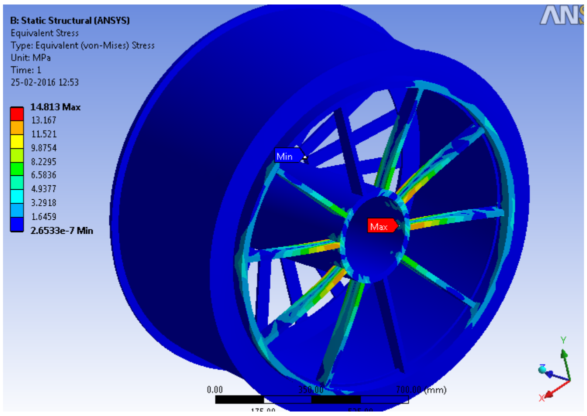

Fig – 7: - Equivalent Stress

6. CONCLUSION

This paper presents the study of the design and optimization of Central Drum in Mine Hoist. As we see the central drum in mine hoist is large and heavy for manufacturing and installation and to rotate full body with loads. The study gives the new design which can reduce the weight of central drum. The design of side disc type structure is replaced with Arm type structure, which makes the central drum little light in weight without affecting its strength. By reducing the weight of central drum, we made the central drum easy to manufacture and installation.

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