RESEARCH ARTICLE OPEN ACCESS

Design and Analysis of Pusher Arm Hub

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Abstract:

Tilting of arm is easily available in existing design as working arm is for shifting the big sheetmetal body from one station to another to get drilled. Now task is to design hub which allows turning of robotic arm along with tilting without affecting its working behavior. In this study design and evaluation of hub body formed by weldment sheet metal components is done. validation of strength is done by carrying out static structural analysis in ANSYS.

Keywords **— Component; hub; weldment; sheetmetal; hub.**

I. INTRODUCTION

Generally a typical arm used for holding and performing operations have plate to plate joints as they are made for only single movement and operation. We are developing arm for making bottom to top machine drilling for object. Object is of height 3.8 meter and mounted vertical on tool set. robotic arm will hold the tool post and drilling tool which work from bottom and end on top. Holding of this arm is quite difficult with the plates or flanges hence we are providing gussets between two flanges to divide forces in cylindrical shape body i.e. hub. So rotation and vertical linear drilling will be easy for this arm. Because of vertical angular tilting perfect holder with new cylindrical hub is necessary here.

Fig. 1. Roller to be drilled by putting it vertically

We want new product development in structure based model on bottom cylindrical body weldment for robotic.

It is needed for the stability , 160 degree rotations and structural strength to hold the arm assembly. Features of drilling arm are, 1) Arm load Capacity force is 3000 N ,
2) Motorised auto tilting and spot picking 3)Drilling application on diffusers body.

This is Completely a new product development in customized assembly arm automation. In this study we will design vertical cylindrical hollow body, flange mounting weldment on both sides, hub with gusseting plates and fasteners joints. Inner diameter of hub is 232 mm and height is 290 mm. After applying the appropriate boundry conditions sresses in the hub will be evaluated

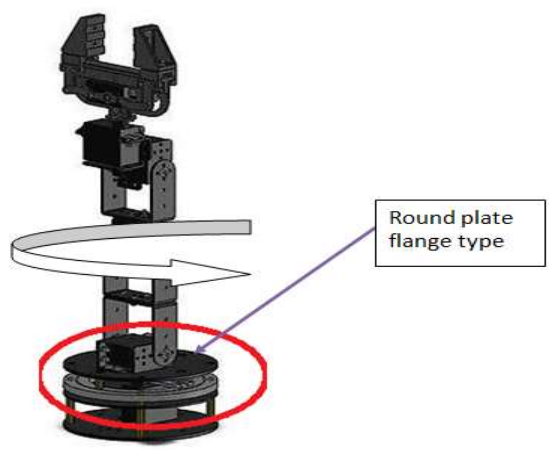


Fig. 2. round flange hub

II. LITERATURE REVIEW

Hui Yang, Jiazhen Hong, Zhengyue Yu [1] has studied characteristics of a flexible hub-beam system with a tip mass in their paper. Based on the Hamilton theory and the finite element discretization method, and in consideration of the second-order coupling quantity of the axial displacement caused by the transverse displacement of the beam, the rigid–flexible coupling dynamic model (referred as the first-order approximation coupling (FOAC) model in their paper) and the corresponding model in non-inertial system for the flexible hub-beam system with a tip mass are presented firstly, then they have studied dynamic characteristics of the system through numerical simulations under twos cases: the large motion of the system is known and is unknown.

To highlight the essential nature behind the vibration phenomena, Zhen Zhao, Caishan Liu et al.[4] have analyzed the steady vibration of a rotating Euler–Bernoulli beam with a quasi-steady-state stretch. They have used Newton′s law is to derive the equations governing the beam′s elastic motion and the hub′s rotation. A combination of these equations results in a nonlinear partial differential equation that fully reflects the mutual interaction between the two kinds of motion. Via the Fourier series expansion within a finite interval of time, we reduce the PDE into an infinite system of a nonlinear ordinary differential equation (ODE) in spatial domain. they further nondimensionalize the ODE and discretize it via a difference method. The frequencies and modal shapes of a general rotating beam are then determined numerically. For a low-speed beam where the ignorance of geometric stiffening is feasible, the beam′s vibration characteristics are solved analytically. Then they have validated their numerical method and the analytical solutions by comparing with either the past experiments or the past numerical findings reported in existing literature. Finally, systematic simulations are performed to demonstrate how the beam′s eigen frequencies vary with the hub′s inertia and rotating speed

In the study done by Reddy and Chin[6], they have investigated the stability of cylindrical shells that composed of ceramic, FGM, and metal layers subjected to axial load and resting on Winkler-Pasternak foundations. Material properties of FGM layer are varied continuously in thickness direction according to a simple power distribution in terms of the ceramic and metal volume fractions. The modified Donnell type stability and compatibility equations on the Pasternak foundation are obtained. Applying Galerkin's method analytic solutions are obtained for the criti-cal axial load of three-layered cylindrical shells containing an FGM layer with and without elastic foundation. The detailed parametric studies are carried out to study the influences of thickness variations of the FGM layer, radius-to-thickness ratio, material composition and material profile index, Winkler and Pasternak foundations on the critical axial load of three-layered cylindrical shells. Comparing results with those in the literature validates the present analysis.

Sofiyev and Avcar[2] has discussed in their paper that under combined electro-thermo-mechanical loadings, the nonlinear bending of piezoelectric cylindrical shell reinforced with boron nitride nanotubes (BNNTs) . By employing nonlinear strains based on Donnell shell theory and utilizing piezoelectric theory including thermal effects, the constitutive relations of the piezoelectric shell reinforced with BNNTs are established. Then the governing equations of the structure are derived through variation principle and resolved by applying the finite difference method. In numerical examples, the effects of geometric nonlinear, voltage, temperature, as well as volume fraction on the deflection and bending moment of axisymmetrical piezoelectric cylindrical shell reinforced with BNNTs are discussed in detail.

Chamat, Touache et.al.[3] has used two cylindrical vessels under internal pressure for their work in order to study the influence of the position and size of defects on their elastic and elastoplastic behavior. One contains two external longitudinal semi-elliptic defects of different dimensions realized diametrically opposed. The other contains the same defects but is circumferential. These defects are carried out by elect-erosion. Strain gauges are placed in the neighborhood of the defects of which the purpose is to obtain the strain distribution. This work also allows the comparison between two defects of different dimensions, which are of the same shape or different shapes. These defects are longitudinal and circumferential semi-elliptical. The position of these defects relative to the inner radius of a cylindrical pressure vessel is considered. The deformations results are then discussed. The experimental study is performed on models consist of a cylindrical shell closed by two semi spherical fund large square radius The external and internal diameters are 406.4 mm and 386.4 mm, respectively

M.N. Hamdan, A.H. El-Sinawi [5] in their work deals with the development of a dynamic model for a slender flexible arm undergoing relatively large planar flexural deformations, and clamped with a setting angle to a compliant rotating hub. The model is derived assuming an inextensible Euler–Bernoulli beam, but takes into account the axial inertia and nonlinear curvature.

III. DESIGN AND DEVELOPEMENT

A. *Hub*

Material used for hub is AISI316. Hub will having base mount arrangement, vertical stability with multiple loading conditions, revolving flange on flange mechanism and customized bushing for long vertical arm bottom.

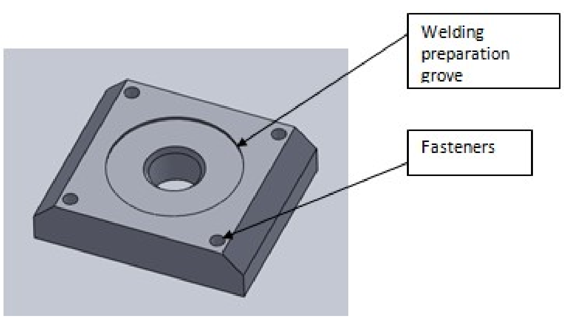


Fig. 3. Base for hub

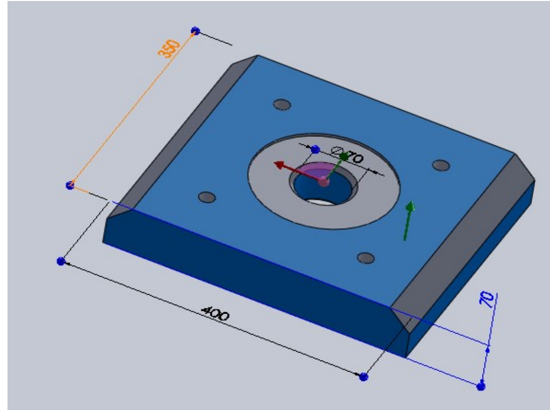


Fig.4. Dimensions of base

. Calculations involved for hub drum are,

Inner diameter (d) = 160mm

Outer diameter (D) = 180 mm

Height(L) = 250mm

Load(W) = 30KN

Maximum deflection will be at free end and given by

$$\delta_{max} = \frac{WL^3}{3EI}$$

Where ,
E = Modulus of Elasticity = 200 GPa
I = Moment of Inertia

$$I = \frac{\pi}{64}(D^4 - d^4)$$

I = 19360064.73 mm^4

$$\boldsymbol{\delta_{max} = 0.04\ mm}$$

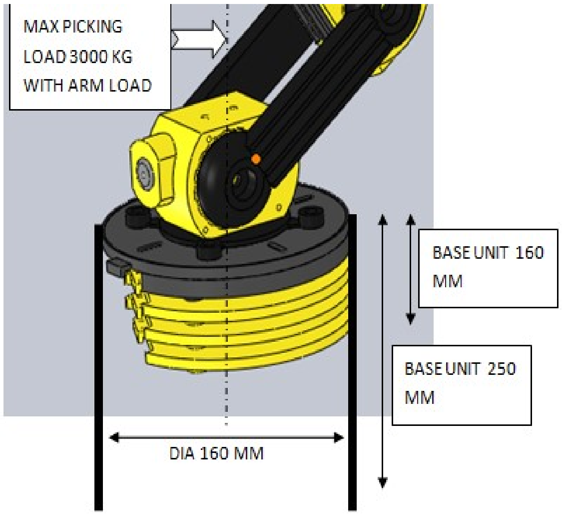
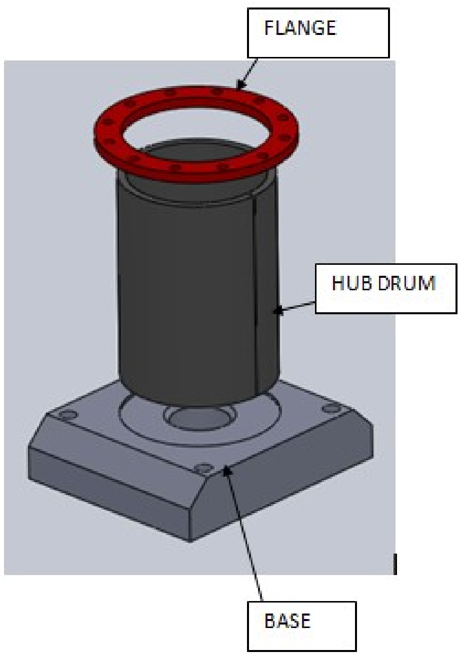


Fig 5 Dimensions for hub



. Fig, 6 Basic Mechanical Components in hub design

Maximum Stress will be at support and is given by

$$\sigma_{max} = \frac{M}{Z}$$

Where,

M = Maximum bending moment

= W x L

= 30000 x 250

M = 7500000 N mm

Z = Section Modulus

$$Z = \frac{\pi}{32}\left(\frac{D^4 - d^4}{D}\right)$$

$$Z = \frac{\pi}{32}\left(\frac{180^4 - 160^4}{180}\right)$$

Z = 215111.83 mm^3

Hence

Maximum Stress

$$\sigma_{max} = \frac{7500000}{215111.83}$$

$$\boldsymbol{\sigma_{max} = 34.87\ N/mm^2}$$

Since we are providing welding joint between drum hub and base mount. In observation it is found that it must be having gusseting to support against arm work bending force. so we will design it.

B. ***Gussete mounting***

Four gussets are capable to hold bending load. when arm is loaded and activated against gusset, hub body load comes on single side and it acts on to bend on arm acting side so we are providing gusset to retrieve the normal condition against load. Now loads are expecting on four sides since arm is to be rotate and worked for 360 degree. So we used 4 gussets

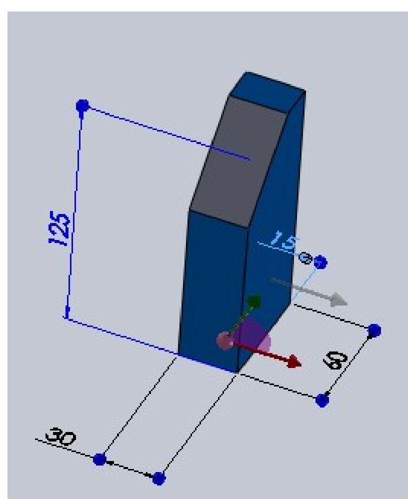


Fig.7. Gussete geometry

Stresses in Gusset

Maximum Total load on the gusset is approximate

30 kN

Maximum deflection will be at free end and given by

$$\delta_{max} = \frac{WL^3}{3EI}$$

Where ,
E = Modulus of Elasticity = 200 GPa
I = Moment of Inertia

$$I = \frac{bd^3}{12}$$

I = 7812.5 mm^4
Hence,

$$\delta_{max} = 12.5\ mm$$

Maximum Stress will be at support and given by,

$$\sigma_{max} = \frac{M}{Z}$$

Where,

M = Maximum bending moment
= WL
= (30000 x 125)
M = 3750000 N mm

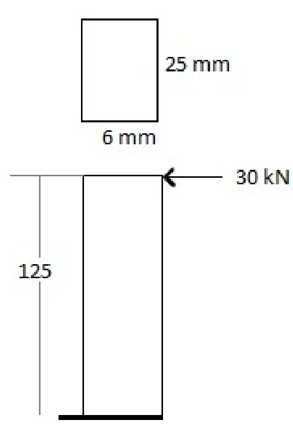


Fig. 8 Dimensions of gusset

Z = Section Modulus
Z = 625 mm^3
Hence,
Maximum Stress

$$\sigma_{max} = \frac{3750000}{625}$$

$$\sigma_{max} = 6000\ N/mm^2$$

This value of maximum stress is too large and is not practical.It is found that the deflection and the stress developed in the gusset are exceeding the safe value. Hence the gussets are not safe and need to redesign.Now as Yeild strength for steel is 215 MPa, the Maximum permissible stress for gusset should be 215 MPa
Hence,

$$215 = \frac{M}{Z}$$

$$215 = \frac{3750000}{Z}$$

$$Z = 17441.86\ mm^3$$

$$Z = \frac{bd^2}{6}$$

d = 60 mm

Hence,

$$17441.86 = \frac{b\ x\ 60^2}{6}$$

b = 29. 06 mm

Hence ,

b = 30 mm and

d = 60 mm

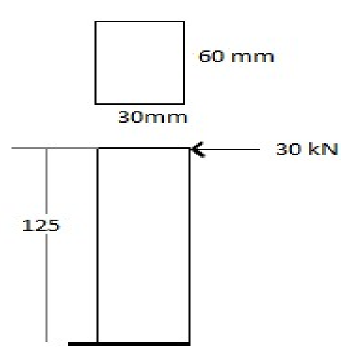


Fig. 9. Dimensions of new guesset

Hence,

Maximum deflection will be at free end and given by $\delta_{max} = \frac{WL^3}{3EI}$

Where ,

E = Modulus of Elasticity = 200 GPa

I = Moment of Inertia

$$I = \frac{bd^3}{12}$$

$I = 540000\ mm^4$

Hence

$\delta_{max} = 0.18\ mm$

And maximum Stress will be,

$$\sigma_{max} = \frac{M}{Z}$$

Where,

M = Maximum bending moment

M = WL

M = (30000 x 125)

M = 3750000 N mm

Z = Section Modulus

$Z = 18000\ mm^3$

Hence maximum Stress,

$$\sigma_{max} = \frac{3750000}{18000}$$

$$\sigma_{max} = 208.33\ N/mm^2$$

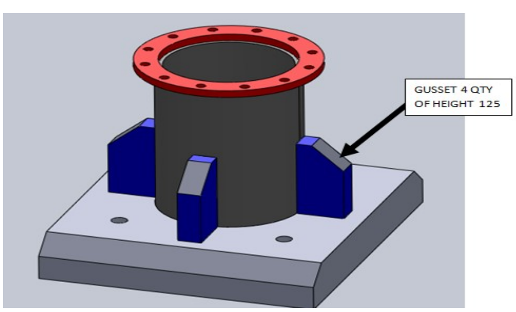


Fig.10. positioning of gussets

IV. CAE VALIDATION

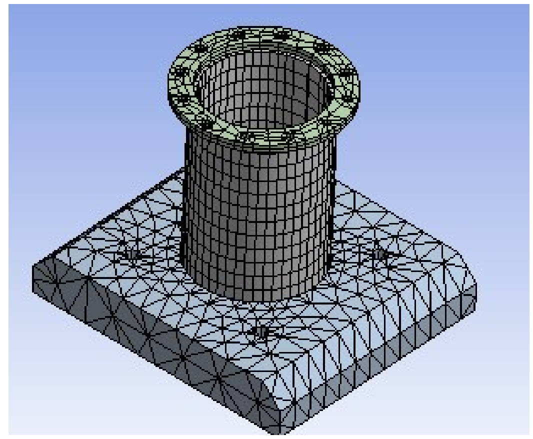


Fig. 11. Meshing body

Table 1. meshing details

Fig. 13. Bending stress found 24 MPA

Details of "Mesh"

Defaults	
Physics Preference	Mechanical
Relevance	0
Sizing	
Use Advanced Si...	Off
Relevance Center	Coarse
Element Size	Default
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Coarse
Minimum Edge L...	10.0 mm
Inflation	
Use Automatic Te...	None
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Inflation Algorit...	Pre
View Advanced ...	No

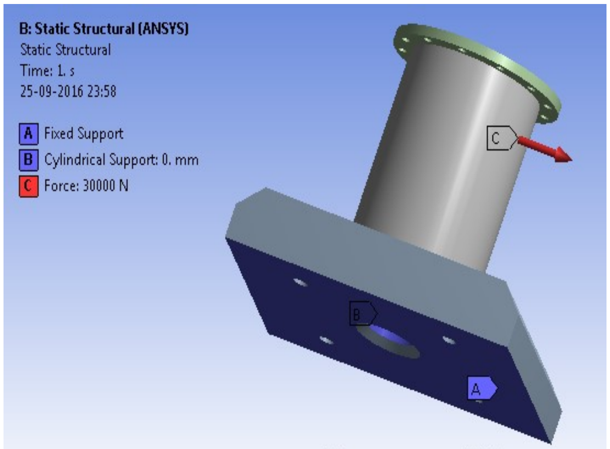


Fig. 11. Loading parameters

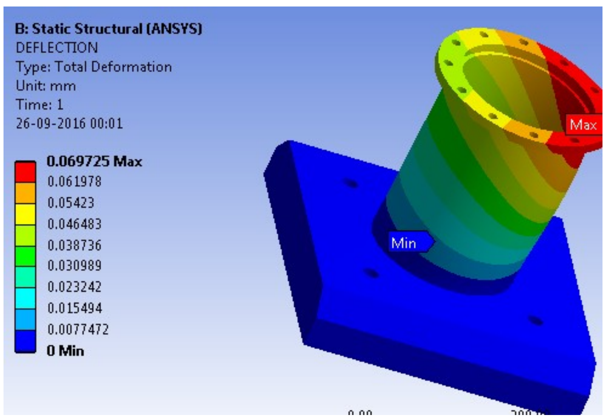


Fig. 12. Deflection max 0.06 mm

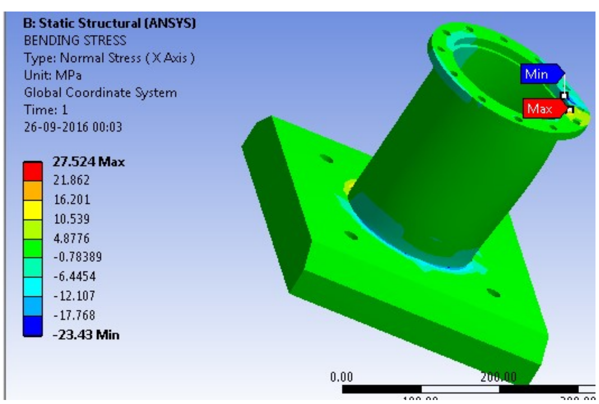


Fig. 13. Bending stress found 24 MPA

Individual loads for guessets are shown below.

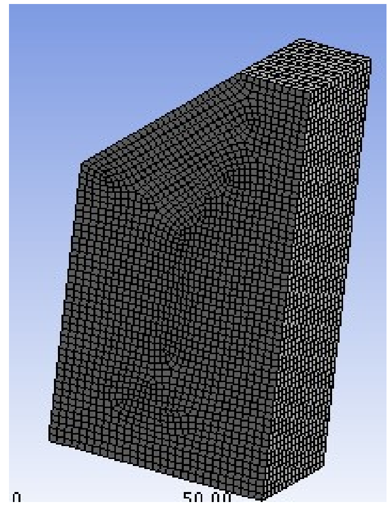


Fig. 14. Meshing model of guessets

Table. 2 Details of meshing (guessets)

Details of "Mesh"	
Defaults	
Physics Preference	Mechanical
Relevance	0
Sizing	
Use Advanced Size Function	Off
Relevance Center	Fine
Element Size	Default
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Coarse
Minimum Edge Length	20.0 mm
Inflation	
Use Automatic Tet Inflation	None
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Inflation Algorithm	Pre
View Advanced Options	No

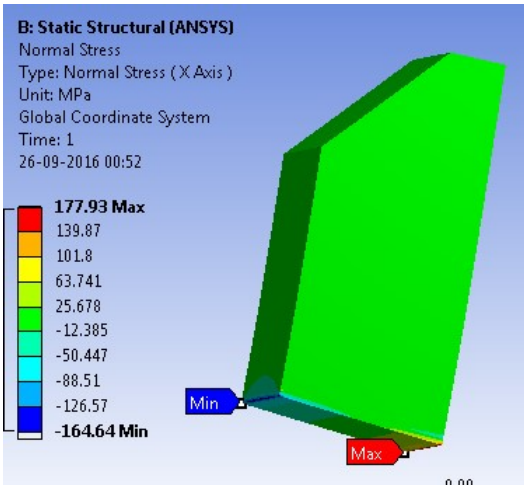


Fig. 15.Stress zones and Max value of stress 178 Mpa

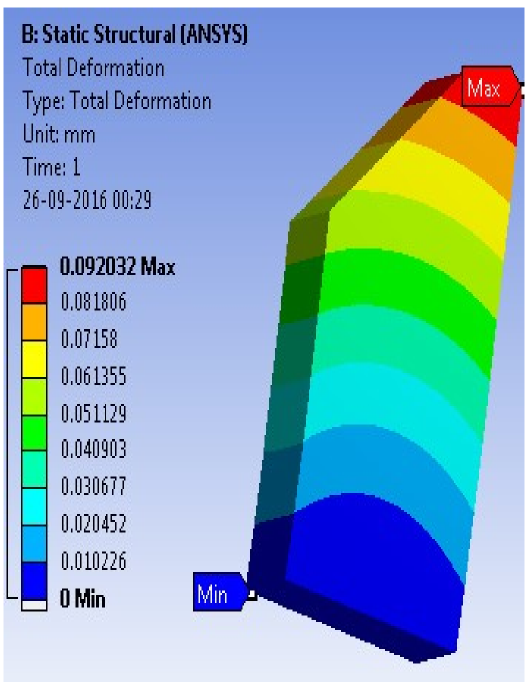


Fig. 16 0.1 mm max deflection found at shown zone

V. CONCLUSION

The results obtained from calculations and CAE validation are tabulated in table ! and Table 2

Table.3 Cylindrical hub

Parameter	Design	CAE
Stress (mpa)	34.87	24
Deflection(mm)	0.04	0.06

Table 4 Results of guessets

Parameters	Design	CAE
Stress(mpa)	208	178
Deflection(mm)	0.18	0.1

From the results obtained from design of hub and guessets and the CAE validation it is found that cylindrical vessel shaped hub can be feasible to make rotary movement and indexing arm for operating and fixing in this hub. Only welding efficiency need to be taken in care

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