

Design, Analysis and Parameterization of Girth Gear

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Abstract—Girth gears are large ring gears that are fitted around the cylindrical mills or kilns and acts as a primary rotational drive of the system. Several standards like AGMA (American Gear Manufacturers Association), ISO (International Organization for Standardization) are available for gear design. In the present work, a detailed way of designing girth gears based on pitting resistance, bending strength, manufacturing feasibility and other practical constraints has been developed using AGMA standard and essential engineering knowledge. The theoretical result obtained is validated with the analysis carried out using SOLIDWORKS software. The model is parameterized and automated model generation database is created using Xpressdrive. The theoretical calculations are tabulated using MS – Excel for easy access.

Index Terms— girth gear, AGMA standards, gear tooth, pitting resistance, bending strength.

I. INTRODUCTION

A rotary kiln is a pyro-processing device used to raise materials to a higher temperature (calcinations) in a continuous process. Large girth gears are used where heavy loads need to be carried out in rotary tubular kilns (pyro processing device) and tube mills. In the present work, the girth gear of a pelletizing drum will be considered for design and analysis.

The major objectives of the project are as follows. The first aim is to create a standard procedure to build a girth gear under AGMA standards, using essential engineering knowledge and also taking practical constraints into consideration. The next aim is to create a 3D model using SOLIDWORKS, parameterize the model and automate it using SOLIDWORKS XPRESSDRIVE software. Then the generated model has to be analysed using SOLIDWORKS. Finally, the GUI has to be prepared in excel format for theoretical calculation. The comparison of the results has to be done and a detailed report has to be generated.

It provides a high starting torque to rotate the shell with large eccentric load and self-weight of the components at very low angular velocity.

Robert Errichello (1995) discusses in depth about the proper approach to design a gear set with optimum load capacity, number of teeth and gear correction. The pitting resistance factor and bending resistance factor relations have been clearly elucidated. The effects of fatigue in the process of gear design are also explored [6].

Venkatesh B., et al (2010) designed a helical gear and modelled it using CATIA. Subsequently analysis using ANSYS were carried out. The modified

Lewis form factor was used for designing the gears. In the analysis, the interactions of gear tooth were clearly presented. The necessary analysis required to validate the result are stated [3].

Krishnakant kankar., et al (2009) have presented a general method of generating profile of spur gear and its specification derived using MATLAB. The coding for parameterization of spur gear model was clearly stated.

The necessary specifications for parameterization were used to create the profile of gear [4].

Zeping Wei (2004) investigated the characteristics of an involute gear system including contact stresses, bending stresses and the transmission errors of gears in mesh. The virtual analyses of gear tooth, during meshing, were examined by using ANSYS software [8].

Anand B. Rathod., et al (2011) have discussed the methodology for creating the parametric modelling of spur gear using ProE software. The model is created with basic specifications of gear. The involute profile of the gear is created using parametric equation of involute profile [2].

Mirzaei M., et al (2014) described the major activities carried out in the process of failure analysis of the girth gear of an industrial ball mill. The influence of lubrication in the working of gears was clearly explored [5].

Amol Deshpande et al (2014) determined the bending and pitting resistance using AGMA standard. They then compared the theoretical results of the ANSYS analysis report. The error was less than 5% upon comparing the both results [1].

Puttaswamaiah S., et al (2012) have performed the finite element analysis of spur gear using ANSYS software and the errors occurring during transmission were evaluated. The detailed analysis was carried out with spur gears with zero addendum modification factors [7].

II. THEORETICAL AND DESIGN CALCULATIONS

A. Static Analysis of Pelletizing Drum Assembly

The Position of girth gear in the setup is decided based on the static load (Dead load) distribution of the components and loaded material. The dead loads include self weight of gear, tyres, flanges, pelletizing drum and refractory lining and material loaded in. It is recommended that the Girth gear should be placed nearer to the support rollers and the inclination of the drum doesn't have significant effect in the load distribution.

B. Gear tooth and pinion tooth - Design Calculation

The Gear tooth of the girth gear is designed based on the AGMA standard. The AGMA standard includes gear tooth design based on bending strength and pitting resistance. Based on the various literature surveys, experience and other parameters, the factor values are assigned. The following AGMA standard were used: (i) ANSI/AGMA 2001-D04 (Revision of ANSI/AGMA 2001-C95) and (ii) AGMA 908-B89 (Revision of AGMA 226.01)

Pitting/contact resistance criteria: Taking safety factor for pitting,

$$S_H = 1.10$$

Contact stress number,

$$S_C = 189.312 \sqrt{\frac{413634.29 * 1.385 * 1.63 * 1.25 * 1.76 * 1.25}{666.67 * 660^{0.251}}} \text{ Pa} \quad (1)$$

$$S_C = 815.35 \text{ MPa}$$

Allowable contact stress number,

$$S_{c,all} = \frac{811.168 * 0.9407 * 1.022}{1.1 * 1 * 0.85} \text{ Pa}$$

$$s_{c,all} = 816.17 \text{ MPa}$$

Comparing s_c and $s_{c,all}$ $s_{c,all}$ is greater than s_c , hence pinion tooth's design is safe based on pitting resistance criteria.

Bending strength criteria: Taking safety factor for bending strength,

$$S_F = 1.10$$

Bending stress number,

$$s_t = \left(413634.29 * 1.25 * 1.385 * 1.63 \frac{1}{660 * m_t} \frac{1.76 * 1}{0.55} \right) \text{ Pa} \quad (2)$$

$$s_t = 223.48 \text{ MPa}$$

Allowable bending stress number,

$$s_{t,all} = \frac{234.818 * 0.9706}{1.1 * 1 * 0.85}$$

$$s_{t,all} = 243.77 \text{ MPa}$$

Comparing s_t and $s_{t,all}$, $s_{t,all}$ is greater than s_t , hence pinion tooth's design is safe based on bending strength criteria.

C. Rim and web - Design Calculation

The Rim contributes the major portion of mass or weight of the girth gear and the web for supporting the rim on the flange. Thus the design of rim has to be done carefully. Considering the safe design, the web structures are assumed to have a minimum mass and standard arm design.

Density of the rim may vary according to the size of the actual girth gear. The area of rim's cross-section is not even and thus an extra material support is provided. Hence, the mass factor $-k_{mass}$ is used. Based on the manufacturing experience (trial and error method), the value of the mass factor is determined. Else, ratio of actual mass to the actual volume is considered for calculating density and k_{mass} value is assigned as unity [3].

The transformation from cylindrical to Cartesian coordinate system can be carried out directly, since the angle of interest,

$$\theta = \frac{360}{4 * Z} \quad (3)$$

$$\begin{Bmatrix} F_x \\ F_y \\ F_z \end{Bmatrix} = \begin{bmatrix} \cos \theta & \sin \theta & 0 \\ -\sin \theta & \cos \theta & 0 \\ 0 & 0 & 1 \end{bmatrix} * \begin{Bmatrix} F_t \\ F_r \\ F_{axial} \end{Bmatrix} \gg \begin{Bmatrix} F_x \\ F_y \\ F_z \end{Bmatrix} = \begin{Bmatrix} F_t \\ F_r \\ F_{axial} \end{Bmatrix} \quad (4)$$

' θ ' value becomes significant only when $Z < 14$, which is not preferred.

Hence, the portion of the rim between arms is considered as a beam that is fixed at both ends and is uniformly loaded.

The moment generated by the axial force (F_{axial}) is highly resisted by the moment of inertia of the gear's cross-section, perpendicular to axial force direction. The influence of axial force due to helical gearing is negligible compared to radial and tangential forces.

The Shape factor is a geometry property and it resists moment. The Gear's cross-section is modelled in such a way that the *shape factor* ≥ 1 .

D. Stress in the rim

Tensile stress due to centrifugal force

$$(\sigma_t) = \sigma_t = k_{mass} * \rho * V^2 = 28000 \text{ Pa} \quad (5)$$

Tensile bending stress caused by the restraint of the arms,

$$\sigma_b = \left[\left(\frac{19.74 * k_{mass} * \rho * V^2 * R}{n^2 * t} \right) + \left(\frac{F_r * \pi * R}{n * b * t^2} \right) \right] \quad (6)$$

$$= 10804099.27 \text{ Pa} = 10.8 \text{ MPa}$$

Total tensile stress in the Rim,

$$\sigma = \left[\left(\frac{3}{4} \sigma_t \right) + \left(\frac{1}{4} \sigma_b \right) \right] = 21000 + 2701024.818 = 2.722 \text{ MPa} \quad (7)$$

The maximum possible stress in the Rim is approximately 11 MPa. Considering factor of safety as '8', the total tensile stress induced in the rim is 88 MPa which is below yield stress of the material [13].

Therefore, the design of rim is considered safe.

E. Stress in the arm

Tensile stress due to centrifugal force,

$$\sigma_{t1} = \frac{3}{4} \sigma_t = 21000 \text{ Pa} \quad (8)$$

Bending stress due to the torque transmitted,

$$\sigma_{b1} = \frac{M}{Z} = \left[\left(\frac{T}{R} \right) + (F_r * \pi * R) \right] * \left(\frac{R-r}{n * Z_{arm}} \right) = 220875425.5 \text{ Pa} \quad (\text{Or})$$

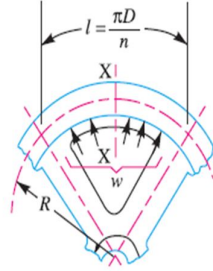


Fig.1. Sectional view of rim

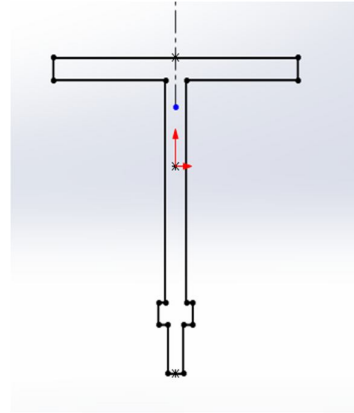
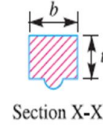


Fig.2. Structural design of girth gear consisting of T-section cross Structure

$$\sigma_{b1} = \frac{M}{Z} = \left[\left(\frac{T}{R} \right) \right] * \left(\frac{R-r}{n * Z_{arm}} \right) = 55424032.05 \text{ Pa} \quad (9)$$

Crushing stress due to the radial force,

$$\sigma_{crush} = \frac{F_r}{A_{arm}} = 4320204.631 \text{ Pa} \quad (10)$$

Total tensile stress in the arms at the flange

$$\sigma = \sigma_{t1} + \sigma_{b1} - \sigma_{crush} = 216.576 \text{ MPa} \quad (11)$$

Considering factor of safety '2', the maximum induced stress is 433.152 MPa which is lesser than yield stress of the material. (Or)

$$\sigma = \sigma_{t1} + \sigma_{b1} - \sigma_{crush} = 51.1248 \text{ MPa} \quad (12)$$

Considering factor of safety '8', the maximum induced stress is 408.998 MPa which is lesser than yield stress of the material. Therefore, the design of rim is again considered safe.

F. Web cross section - Design Calculation

The Cross-section is assumed to be simply supported beam or cantilever beam or any other simple structure for easy calculation. The uniform distributed load (UDL) is assumed to be loaded on the beam considered and proper boundary conditions are provided. The load considered as UDL is self-weight between web ribs i.e. rim portion along the web [12]. A liberate factor of safety is assumed in order to avoid radial load and tangential load influence the web cross-section. The actual deflection can be taken as 100 times greater than or equal to obtained deflection [6].

Overhung Deflection = $\delta = 5.909e - 3 \text{ mm}$. Actual overhung deflection is 0.59 mm lesser than the allowable deflection. Hence, design is safe.

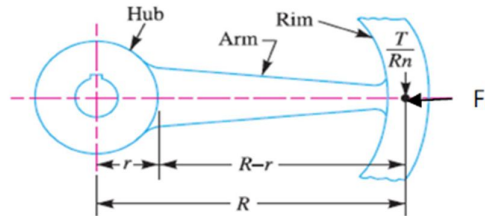


Fig.3. Cantilever beam like arm (web)

G. Shape Factor

The equivalent area axis is found to be lying 64.2325 mm below the outer fibre of the rim. Based on the minimum cross-sectional area condition,

Total area, $A = 93217.6 \text{ mm}^2$

Plastic section modulus, $Z_p = 6472091 \text{ mm}^3$

Elastic section modulus, $(Z_e)_{\text{top}} = 25290015.52 \text{ mm}^3$

Elastic section modulus, $(Z_e)_{\text{bottom}} = 5398584.296 \text{ mm}^3$

$$\text{Shape factor} = \frac{(Z_p)}{(Z_e)} = 0.2559 \quad \text{- top layer} \quad (13)$$

$$\text{Shape factor} = \frac{(Z_p)}{(Z_e)} = 1.1988 \quad \text{- bottom layer} \quad (14)$$

Thus the girth gear is safely designed based on AGMA and engineering knowledge.

III. MODELLING, MESHING AND ANALYSIS

The girth gear and pinion are modelled as single part respectively, to avoid problems during virtual analysis. The gear teeth profile follows involute profile which is drawn via parametric equation of the involute curve [4]. The various dimensions of the gear are parametrized during modelling itself i.e. variables are assigned. The model of gear and pinion are meshed in 3D elements. The element size is determined by convergence test. The elements in the gear teeth are increased to improve the accuracy [8]. The gear can be meshed easily in hypermesh and solidworks as a whole part. But gear is considered as segment to avoid computation problem in hypermesh. The element size can be increased in solidworks to mesh as a whole part.

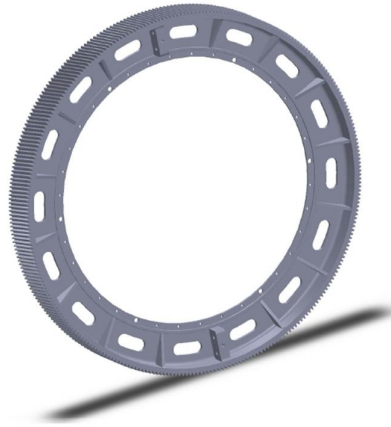


Fig.4. Model of girth gear assembly designed in SOLIDWORKS

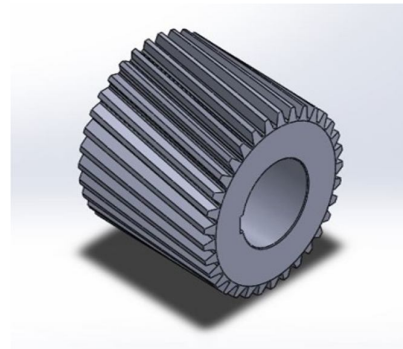


Fig.5. Model of pinion designed in SOLIDWORKS

Thus the models are meshed using curvature based mesh in SOLIDWORKS and tetrahedron mesh in HYPERMESH.



Fig.6. Meshed model of girth gear

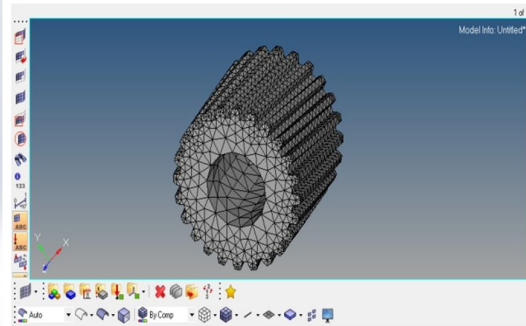


Fig.7. Meshed model of pinion in HYPERMESH

CONDITION 1:

The gear is considered to be fixed in all dof along the flange end. Loads are applied along the pitch line in a single teeth.

In static analysis, the maximum bending strength is obtained by considering condition 1. The Maximum Von-Mises stress of 209.8 MPa is found to be occurred along the pitch line with maximum deflection of 0.6794 mm.

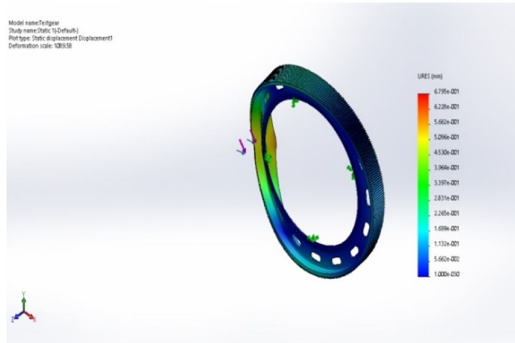


Fig.8. Displacement plot of girth gear model for condition 1

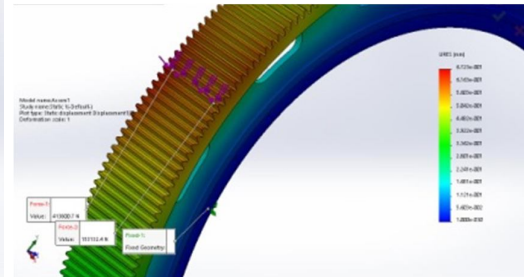


Fig.9. Loading condition and stress plot for condition 1

CONDITION 2:

The gear's center is coupled with the flange end using link connector i.e. the load applied at the center transferred to the flange-end. All dof except z-axis rotation are arrested [10].

The pinion is meshed with the gear based on contact ratio and center distance. And it considered to be rigid body with all dof arrested. The torque is applied at the center of the gear [11].

In static analysis, the maximum pitting resistance of 809.4 MPa is obtained by considering condition 2.

Thus, based on the customer requirement, analysis is carried out in Abaqus or solidworks as per accuracy level.

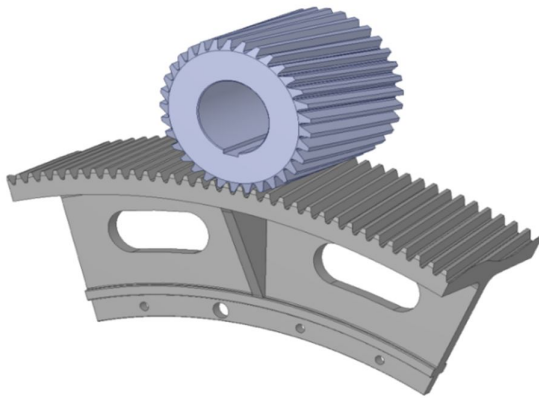


Fig.10. Meshed model before analysis under condition 2

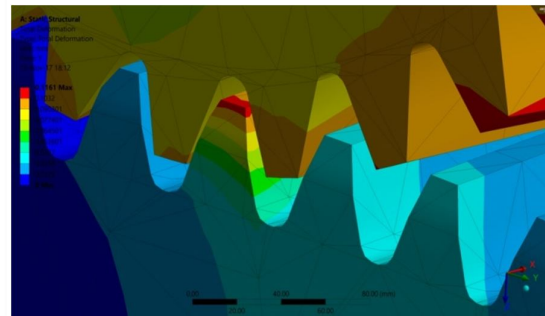


Fig.11. Stress plot of the meshed gear

IV. PARAMETRIZATION AND AUTOMATION

The models are automated using the SOLIDWORKS XPRESSDRIVE software. By providing the data in the data spaces, the model will be generated based on the pre-described rules [9].

V. GUI FOR CALCULATION IN MS-EXCEL FORMAT

The graphical user interface for AGMA calculations for the girth gear has been designed. This interface gives solutions for any critical problems within few clicks. Also the gear geometry specifications have been made visible for the users, as shown in Fig.15.

Initial input data for pinion and gear design will be collected from the user. Accordingly the material data will be calculated. Finally based on this, the data will be computed for finding the gear ratio, pitch line velocity, transverse diametric pitch, transmitted load, axial load, variable load, bending strength and pitching resistance.

FUTURE WORK

The optimized model of the existing design is to be modelled, analysed and compared. The solid-fluid interaction analysis between the gear teeth and lubrication needs to be conducted. The fatigue analyses of the working gears needs to be completed.

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