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DESIGN AND OPTIMIZATION OF ROTARY TURRET PLATE OF POUCHER MACHINE

Jigar G. Patel
Institute of Technology, Nirma University, Ahmedabad – 382481, India
Email:14mmcc17@nirmuni.ac.in

Mitesh B. Panchal
Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad – 382481, IndiaEmail: mitesh.panchal@nirmauni.ac.in

***Abstract* — This paper present design and optimization of rotary turret plate of 5 kg capacity for food product packaging machine. Two different criteria are considered for design; first one is centrifugal force approach with static structural analysis of turret plate and second is with transient structural analysis. A 3D CAD model of rotary turret assembly has been prepared using solid modeling packages. The finite element analysis (FEA) of turret plate has been carried out using analysis software. Consideration of centrifugal force is one of the criteria to analyze the performance and behavior of component in working condition. The moment is applied at the central axis of turret and friction less support is applied on inner surface, where shaft is being attached. The boundary conditions as fixed support has been considered at the different sixteen faces, where bolts have been attached. Centrifugal forces are applied at different eight stations of turret by considering X, Y and Z components of forces. The obtained simulation results for induced stress, deformation and strain depict that the design of rotary turret plate is well within the allowable stress limits of considered material. The strength based and mass based optimization of turret plate has been carried out. The optimized design can be considered as must suitable to rotary turret plate to perform its function efficiently.**

Keywords— Design, Optimization, finite element analysis, Poucher Machine.

I. INTRODUCTION

Turret plate is a rotating component of the poucher machine. The rotation of this plate is intermittent. There is a servo motor mounted which is coupled with a reduction gearbox and both together are mounted on a cylindrical column. The turret plate is coupled with a coupler plate to the gearbox coupling plate. The servo motor drives the turret rotary plate. The pouch to the turret plate is 25.4 mm thick. It is having very high factor of safety and can be optimized for weight criteria. Main concern of this paper is the development of turret of rotary Poucher machine to handle pouches of 5 kg capacity of food products packaging. For that finite element analysis and optimization of turret plate has been carried out to check the strength of turret as main component of Poucher machine.

II. 3-DIMENSSIONAL CAD MODELLING OF ROTARY TURRET ASSEMBLY

The rotary turret assembly indexes the pouch clamps through eight stations. The turret clamp plate is positioned over the stationary oscillating ring cam. It rotates on a centre hub that mounts to the gearbox and is connected to the support column. The gearbox is powered by a servo motor.

Figure 1 shows, the 3D CAD model created using CRE-O of the rotary turret plate assembly including pouch clamps at all eight stations. During each machine cycle, the turret plate rotates one position indexing the pouch clamps through the stations. The pouch clamps carry the pouches to eight index stations around the turret. Each pouch will be opened, filled, closed, top sealed and discharged at these stations. The pouches are removed from the pouch clamps and placed on a discharge conveyor. The cam follower extends from each clamps set through and indentation in the turret plate. It rides against the oscillating ring cam and causes the clamps to move, or pitch, closer together or further apart.

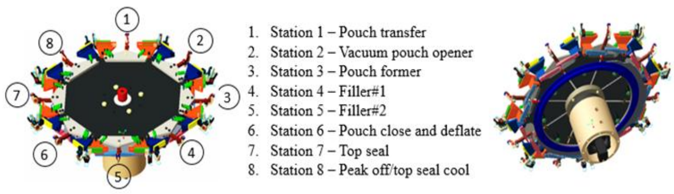

Fig. 1. Rotary Turret Assembly, Isometric View

III. FINITE ELEMENT ANALYSIS OF ROTARY TURRET PLATE

The rotation of turret plate is intermittent. The servo motor drives the turret rotary plate. The thickness of the turret plate is 25.4 mm thick. The material for turret plate is considered as Aluminium T6061 properties as: 7.310e+10 (Young Modulus (N.m−2)), 0.33 (Poisson Ratio), 2710 (Density (Kg.m−3)), 2.36e-5 (Thermal Expansion (K)), 2.41e+8 (Yield Strength (N.m−2)).

Calculating induced stresses inside component by application of centrifugal force and weight of the pouches is one of the method of analysing the performance and behaviour of component in its working condition.

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In this method the turret plate clamp is fixed at the PCD holes of turret plate, where a centre hub couples the gearbox with the turret plate. And the force components i.e. component forces in X, Y and Z direction are applied at the location of mounting of the pouch clamp mounting bracket on the turret plate. The location of the fixed geometry and the orientation and attachment of the centrifugal forces on rotary turret plate is shown in the figure below. Here along with the centrifugal stresses in X, Y direction the weight of the pouch in Z direction is also considered. The centrifugal force on each pouch mass has been calculated manually from the motor specification.

Frictionless support is applied at inner surface of turret plate and fixed support is applied at PCD holes of turret plate, where turret plate is clamped with centre hub. More over the moment is applied at the centre portion of turret plate in clockwise direction and magnitude of moment is 48.6 N-m which is peak stall torque taken from servo motor specification catalogue. There are two types of analysis has been carried out, first is static structural analysis and second is transient structural analysis.

A. *Centrifugal force calculations*

n=114 rpm (outcome rpm of gearbox)

$$\omega = 2\pi n/60 = 11.938 \text{ rad/s}$$

m= 5 kg (mass of one pouch)

r= 0.417 m (distance of clamped pouch from center of turret)

$$F = mr\omega^2 = 300.17 \text{ N}$$

B. *Centrifugal force components*

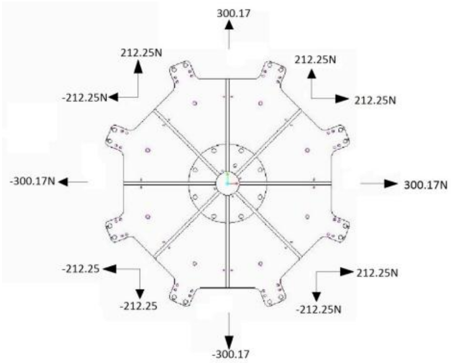


Fig. 2. X ,Y components of centrifugal force at eight different stations

Force 300.17N has two components in X and Y direction. Here eight stations are equally spaced in 360°. So component of this force in X direction is Fcos45°(212.25N). Same as component of this force in Y direction is Fsin45°(212.25N). Distribution of these components of forces is shown in above figure. Here minus sign is only indicating the direction of force, it will not affact the magnitude of force.

Z component is taken 50N in downward direction (-Z direction) which is indicating mass of pouch. Here the mass of pouch is 5 kg, equivalent to 50N.

TABLE 1. X, Y, Z components of centrifugal force

	X[N]	Y[N]	Z[N]
Force 1	300.17	0	-50
Force 2	212.25	212.25	-50
Force 3	0	300.17	-50
Force 4	-212.25	212.25	-50
Force 5	-300.17	0	-50
Force 6	-212.25	-212.25	-50
Force 7	0	-300.17	-50
Force 8	212.25	-212.25	-50

C. *Static structural analysis*

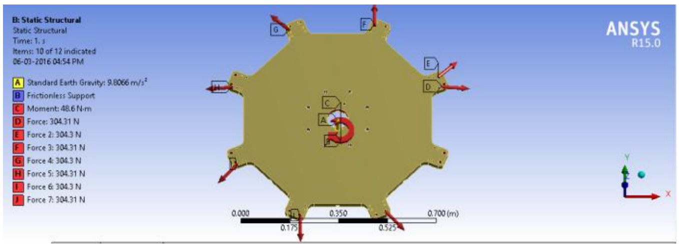


Fig. 3. Analysis setup for centrifugal force approach, static structural

The figure 3. shows the analysis setup for static structural analysis of rotary turret plate.

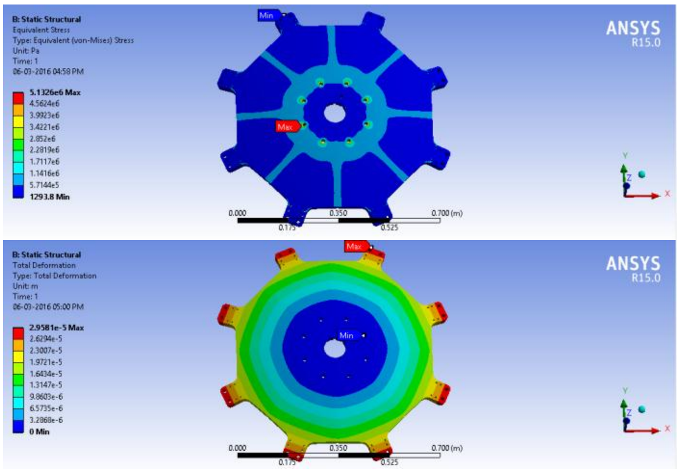


Fig. 4. Analysis results for static structural – von mises stress , total deformation

TABLE 2. Resultant values of Analysis

	Equivalent stress	Total Deformation
Minimum	1293.8 Pa	0 m
Maximum	5.1326 e006 Pa	2.9581e-005 m

The distribution of von-mices (equivalent) stresses over the turret plate using finie element method based simulation approach is shown in Figure 4. From the obtained results we can observe that the maximum localised surface stresses are produced at the inner surface of PCD holes, where turret plate is assembled with rotating mechanism using fastening bolts. The magnitude of this induced localized surface stress is well within the permissible limit of stresses for the considered material. Figure shows the overall deformation of turret plate under considered boundary conditions based on centrifugal forces. The obtained results indicate the maximum deflection of turret plate of magnitude 2.9581e-005 m and corresponding stress value of 5.1326 e006 Pa.

D. Transient structural analysis

Here in transient structural analysis moment is applied in 10 fractions of one second. In which initially the moment in first fraction is considered as 0 N-m, and then gradually it will increase and attain peak value of moment which is 48.6 N-m and then gradually it will decrease and again come to at zero value.

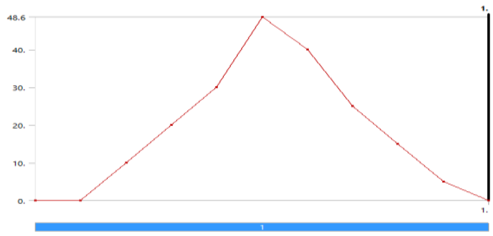


Fig. 5. Moment diagram for transient structural analysis

TABLE 3. Applied moment for transient structural analysis

Steps	Time [s]	Moment [N·m]
1	0.	0
	0.1	
	0.2	10
	0.3	20
	0.4	30
	0.5	48.6
	0.6	40
	0.7	25
	0.8	15
	0.9	5
	1.	0.

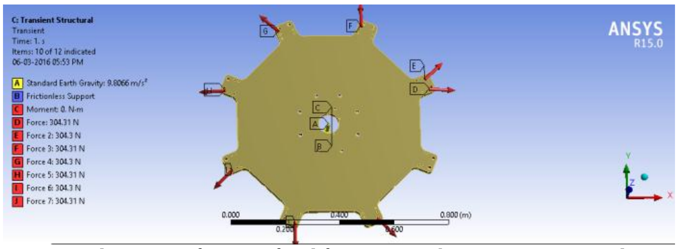


Fig. 6. Analysis setup for centrifugal force approach, Transient structural

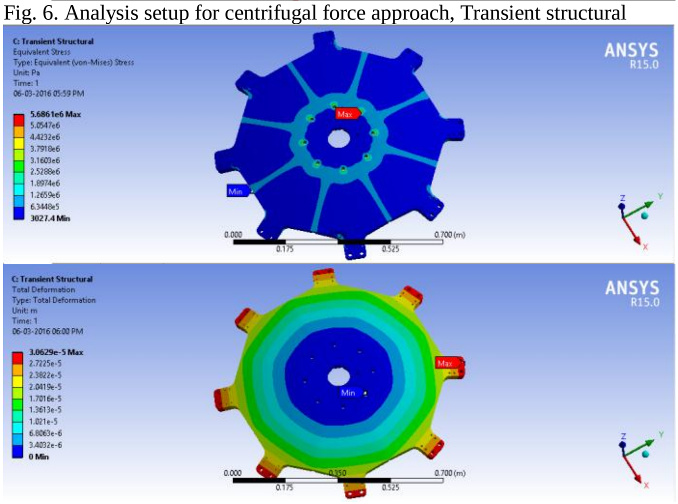


Fig. 7. Analysis results for transient structural – von mises stress , total deformation

TABLE 4. Resultant values of Analysis

	Equivalent stress	Total Deformation
Minimum	3027.4 Pa	0 m
Maximum	5.6861e006 Pa	3.0629e-005 m

From the obtained results we can observe that the maximum localised surface stresses are produced at the inner surface of PCD holes, where turret plate is assembled with rotating mechanism using fastening bolts same as in static structural analysis. The magnitude of this induced localized surface stress is well within the permissible limit of stresses for the considered material. So, large scope of optimization is available there Figure shows the overall deformation of turret plate under considered boundary conditions based on centrifugal forces. The obtained results indicate the maximum deflection of turret plate of magitude 3.0629e-005 m and corresponding stress value of 5.6861e006 Pa.

IV. Thickness Optimization of Rotary Turret Plate

A. Thickness Optimization by DOE

For thickness optimization ANSYS software package tool has been used. The model of the turret plate has been imported and mid-surface of the model in the thickness direction has been generated and analysis has been carried out. Once ANSYS results has been calculated the maximum stress and deformation is set to parameterize for the output requirement and thickness of the plate as input.

	A	B	C	D
1	ID	Parameter Name	Value	Unit
2	Input Parameters			
3	Static Structural (B1)			
4	P4	98078-6M_MIDSURF Thickness	0.0254	m
*	New input parameter	New name	New expression	
6	Output Parameters			
7	Static Structural (B1)			
8	P2	Equivalent Stress Maximum	5.1027	MPa
9	P3	Total Deformation Maximum	0.025938	mm
*	New output parameter		New expression	
11	Charts			

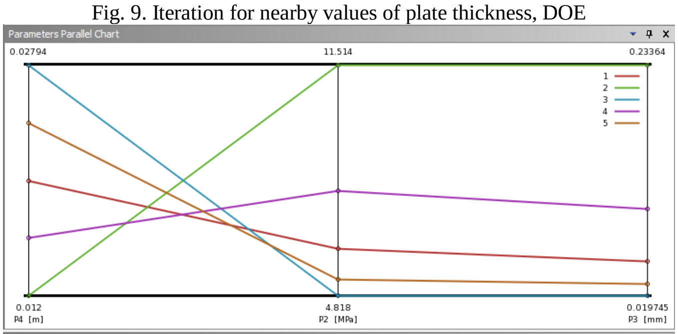
Fig. 8. Parameter settings for DOE tool

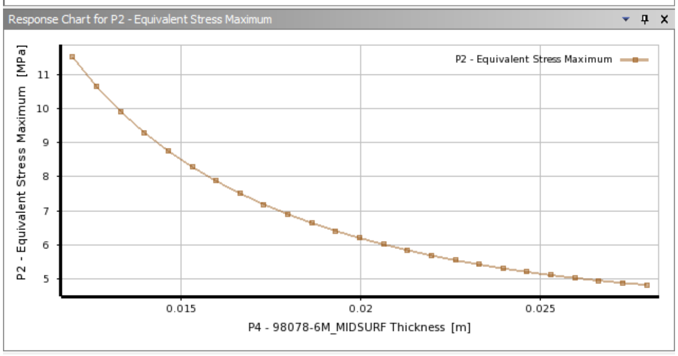
After analysis setup is completed, the data is updated. In analysis setup upper and lower bound value for the thickness parameter is provided. Figure 8. Shows the parameter setup environment of ANSYS optimization module.

Table of Schematic C2: Design of Experiments (Central Composite Design : Auto Defined)

	A	B	C	D
1	Name	P4 - 98078 -6M_MIDSURF Thickness (m)	P2 - Equivalent Stress Maximum (MPa)	P3 - Total Deformation Maximum (mm)
2	1	0.01997	6.1965	0.052167
3	2	0.012	11.514	0.23364
4	3	0.02794	4.818	0.019745
5	4	0.015985	7.8706	0.10024
6	5	0.023955	5.3024	0.030717

Fig. 9. Iteration for nearby values of plate thickness, DOE





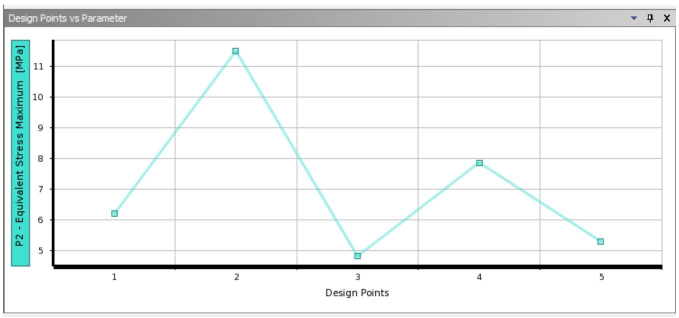


Fig. 10. Output graphs generated for optimization of thickness

Looking to the generated output values of the optimization problem and aluminium plate supplier it has been decided that 16 mm thickness will be a sufficient and suitable thickness for the proposed application. Figure 10 shows the graph of variation in stress corresponding to change in thickness.

B. Thickness optimization by Direct Optimization tool

Properties of Outline A4: P2 <= 150 MPa

	A	B
1	Property	Value
2	General	
3	Parameter	P2 - Equivalent Stress Maximum
4	Objective Name	P2 <= 150 MPa
5	Objective	
6	Type	No Objective
7	Constraint	
8	Type	Values <= Upper Bound
9	Upper Bound	150
10	Decision Support Process	
11	Constraint Importance	Default
12	Constraint Handling	Strict
13	Parameter Details	
14	Units	MPa
15	Calculated Minimum	4.818
16	Calculated Maximum	11.514

Properties of Outline A5: P3 <= 3 mm

	A	B
1	Property	Value
2	General	
3	Parameter	P3 - Total Deformation Maximum
4	Objective Name	P3 <= 3 mm
5	Objective	
6	Type	No Objective
7	Constraint	
8	Type	Values <= Upper Bound
9	Upper Bound	3
10	Decision Support Process	
11	Constraint Importance	Default
12	Constraint Handling	Strict
13	Parameter Details	
14	Units	mm
15	Calculated Minimum	0.019745
16	Calculated Maximum	0.23364

Properties of Outline A6: Minimize P4

	A	B
1	Property	Value
2	⊟ General	
3	Parameter	P4 - 98078-6M_MIDSURF Thickness
4	Objective Name	Minimize P4
5	⊟ Objective	
6	Type	Minimize
7	⊟ Decision Support Process	
8	Objective Importance	Default
9	⊟ Parameter Details	
10	Units	m

Fig. 11. Parameters settings for objectives and constraints, Direct optimization tool

In direct optimization tool of ANSYS objective and constrains of different parameters can be set manually unlike DOE tool in which automatically constrains are being set. In this research work three parameters have been set. In figure 11. Like first parameter equivalent maximum stress has given the constrain that its value will not be greater than 150 MPa. Second parameter total deformation has given the constrain that its value will not be greater than 3mm. And the third one parameter thickness has given the objective like it will be miminiza as possible.

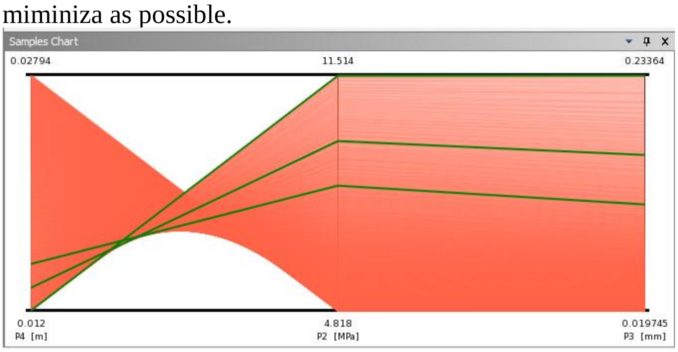


Fig. 12. Sample chart for direct optimization tool

Figure 12. Shows the three candidate points selected by direct optimization tool in which lower one line is look like optimized line.

V. CONCLUTIONS

The presented strength based analysis of rotary turret plate considering centrifugal force due to weight of filled pouches and considered boundary conditions has been found to be safe, as obtained finite element based simulation results are well within the permissible limit of the material considered for turret plate Aluminium T 6061 . The performed analysis also reveal that the thickness of turret plate 25.4 mm, which is used for the 5 kg weight of pouch has very high factor of safety. This analysis has been further extended for the mass based optimization of the rotary turret plate. By thickness optimization process on two different ways it shows that 16 mm thickness is safe for 5 kg of pouch.

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