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Comparison Studies of Sealing Cup Material for Instrumented Pipeline Inspection Gauge

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Abstract

This paper about the material selection or a comparative studies, to selection of material of sealing cup of instrumented pipeline inspection gauge which is used to inspect pipeline in oil/gas industry upto 200 -300 km long without any shutdown of transportation of hydrocarban. These pipelines are buried within the soil upto a depth of 1.5m – 2.5 m traversing various terrains with varying soil resistivity across the world. These pipelines sometimes pass through thickly populated areas carrying highly inflammable products. Regular monitoring of the condition of these pipelines is necessary not only to ensure public health and safety but also to avoid costly and uneconomical shutdowns. The IPIG tools use polyurethane sealing cups for sealing which drives the tool through the pipeline using the fluid pressure. The design of the sealing cup governs the ultimate performance of the tool in terms of providing cutting sealing, resistance to cup wear, buckling, resistance to nose diving. Any abnormal behavior of the sealing cup can result in stuck IPIG within the pipeline calling for emergency shutdown and line repairs.In this studiy we use finite element method for analysis of stress, strain & wear resistance of sealing cup of same dimension with different grade of polyurethen (PU-650 & PU-652).

Keywords: *Polyurethen, IPIG, Sealing Cup, Hydrocarban Pipeline, FEM, Wear Resistance.*

1. Introduction

PIPELINE is efficient method of transport a fluid. These pipeline are buried within the soil upto a depth of 1.5-2.5 meters traversing various terrains with varying soil resistivity across the country In geographical region of any contrary, the hydrocarbon industries operate more than 30000 kms of pipelines for transport crude and oil products across the country.

These pipelines sometimes pass through thickly populated areas carrying highly inflammable products. Regular monitoring of the condition of these pipelines is necessary not only to ensure public health and safety but also to avoid costly and uneconomical shutdowns. Moreover, increased incidences of pilferage of the hydrocarbon products from pipelines pose the danger of fire hazards.

An instrumented Pig (IPIG) is a device for inline inspection of buried pipelines to monitor their health and assess the risk associated with their operation. The IPIG tools use polyurethane sealing cups for sealing which drives the tool through the pipeline using the fluid presure. The design of the sealing cup governs the ultimate performance of the tool in terms of providing cutting sealing, resistance to cup wear, buckling and resistance to nose diving etc. for the expected inspection intervals of the order of 200– 300 kms. Any abnormal behaviour of the sealing cup can result in stuck IPIG within the pipeline calling for emergency shutdown and line repairs.

The design of cups usually has nominal diameters larger than the pipe internal diameter. It has been observed that radial and friction contact forces on oversized discs are high enough to make the cups buckle. Highly oversized sealing cups are at risk to being pulled out of bolt holes on the pig. The too much buckling of cup must be avoided by proper selection of seal geometry and flange diameter. The other failure mode involves the erosion of the outer peripheral edges of the cups to the point where they will no longer provide the requisite sealing. In case of excessive wear out, the tool gets stuck with pressure build up behind the seal. This can initiate tearing at the bolt holes as shown in In order to have longivity of the PU cups in service as well as to overcome the likely failures as mentioned above, it is desired to study an existing design of sealing cup for a 12” diameter IPIG tool for the impact of material charecteristics of two different polyurethane grades, effect of differtial pressure across the cup as well as identify critical regions of failures through Finite element modelling.

2. Literture Review

2.1 Indroduction

Design and control of pig operations through pipelines is discussed by Nieckele et al. [1]. The pig velocity is directly related to the sealing efficiency of the pig, and requires that the liquid flow rate be maintained within certain limits. The flow rate and the pressure distribution depend directly on the profile. The operational design should also account for the pressure distribution along the pipeline, in order to guarantee the level of operating pressure in the pipeline, avoiding the occurrence of either slack flow or excess pressure. He proposed that uneven wear on one side of a cup is usually caused by the pig nosing down during its run due to high line pressure pushing the pig. In continuation with this concept, simulation of the transient motion of pigs through liquid and gas pipelines is presented by Braga et al. [2].

The state of the pipe's internal wall will affect friction. The wall of a typical processed stainless steel pipe is normally very smooth. However the oil industry usually deals with rougher surfaced pipe unless it is lined. The resulting roughness can be in the region of 40 to 150 microns peak to peak. Mathews and Rendle (1994) investigated the wear on multi diameter pigs in 10" flow loop facility. They concluded that less wear was recorded at higher velocities. This is again probably a lubrication effect. In addition, polyurethane as expected wore less than nitrile. That is an initial high rate of wear for the first five kilometers approximately.

The contact forces developed by disk pigs and the pipe wall were predicted by a post buckling finite element analysis of the discs. M. Lesani et al. [3] explained dynamic analysis of small pig through two and three- dimensional liquid pipeline. The derivation and solution of the two and three dimensional dynamic equations for a small pipeline inspection gauge (PIG) through a liquid pipeline was the main aim of this work. The simulation results show that the derived equations are valid and effective for online estimating of the position, velocity and forces acting on the pig at any time of its motion. It is not really known how geometry affects the pig steady state motion, especially small changes from disc to disc. Moore (1972) identifies three direct mechanisms of wear when an elastomer slides on rigid surface:-

- Abrasive wear as a result of sahrp texture in the base surface causing abrasion and tearing of the sliding elastomer.
- Fatigue wear due to blunt base surface projection
- Roll formation in high elastic material sliding on smooth surface

Abrasive wear and fatigue wear result from sliding on rough surface. Roll formation is characterised by high friction sliding on smooth surface. Also fatigue wear is also very mild. For this reason, only abrasive wear will be considered in pig seal wear problem (although very high abrasive pressure pigs could suffer from adhesive wear).

Halling (1978) describes a relationship between distance travelled (X), load (W) and volume of material (Vwear) lost due to wear:-

$$V_{wear} = K_{wear} W X$$

After the pig travel a distance ∂X, material is removed from the pig cups due to wear. The volume of material removed depends on the load supported by the asperities, the distance travelled and the abrasion resistance of the material . The classical relationship used is:-

$$V_{wear} = K_{wear}\, w_a\, \pi\, d\, X$$

Where wa is load supported by the asperities (N/m), d is diameter of pipe and Kwear is coefficient of wear (m2/N)

Figueiredo et al. [4] has presented pig movement in two-phase gas pipelines. Both mechanical and hydrodynamic friction forces occur at the interface of the pig and the pipe wall. The efficiency of pigging operations relies on the velocity with which the pig travels along the line, which in turn is affected by the fluid flow conditions and vice-versa, it becomes crucial to properly predict the pig movement and its hydraulic consequences within the pipeline. Kohda et al. (1988) were pioneer in modelling pig motion in two-phase flows pipelines. The authors presented a model in which the drift-flux model was used to describe the flow in the pipeline and the pig was treated as a moving boundary with the presence of slug ahead of it. The pigging review aims to discover what is already known in the industry regarding the motion of pipeline pigs. Additionally, evidence is provided that a lubrication analysis is the correct way forward in modeling the motion characteristics. Due to absence of design information, it is apparent that even the most basic questions regarding pig differential pressure, leakage and other parameters can only be guessed at. The ability of the pig to perform its function effectively is an even greater challenge.

2.2 Stedy State Motion Model of Pig

In general terms, a pig is a solid cylindrical plug driven through the pipeline by the flowing fluid. Contact forces between the pig and the pipe wall are developed due to the oversize of the pig and should be overcome by the driving pressure provided by the flow. In order to produce efficient sealing, conical cups normally have nominal diameters larger than the pipe internal diameter. Some pigs have passages in the rear cup or in the body to allow some fluid flow through the pig. This bypass flow is used to control the pig velocity.

When the pig is stationary there is an interference fit between pig's sealing cup and the pipe wall (Fig 1). It causes a frictional resistance to pig motion. On start-up, higher pressure acts to destroy this seal to a certain extent. The less pressure required to initiate the pig the better as this is potential energy which is converted to kinetic energy later on during motion. This is the underlying cause of velocity excursion. On acceleration, pig differential pressure drops with increasing velocity.

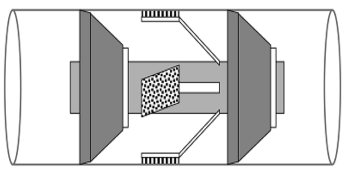

Figure 1: Schematic view of 2-cup magnet module

In Fig. 2, different forces acting on conical cup model in pipeline are shown. The friction force is acting opposite to the direction of motion of pig. The body weight of pig is acting downward. The required propelling force to drive the pig is given by differential pressure across the pig seal. Due to oversized cup, a wall force acts on the outer periphery of the sealing cup.

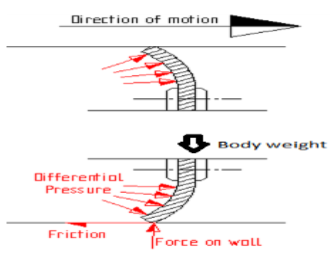

Figure 2: Force diagram of sealing cup

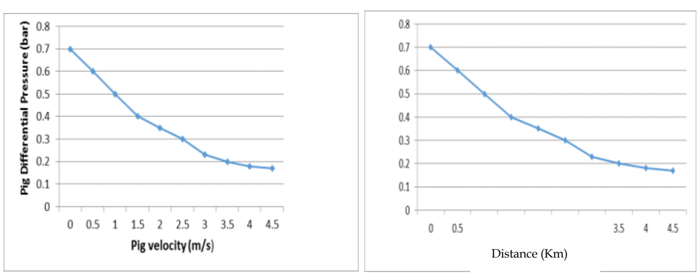

Figure 3: Plot of Pig Differential Pressure Vs Pig Velocity & Distance

2.3 Groverning Equation

$$m\frac{dv}{dt} = (P1 - P2)A - mg\ sin\beta - F \quad \text{------ (1)}$$

FD = 0.5 CD A ρ(Vf – V)2 ------(2)

FD = F=Ff = η N --------(3)

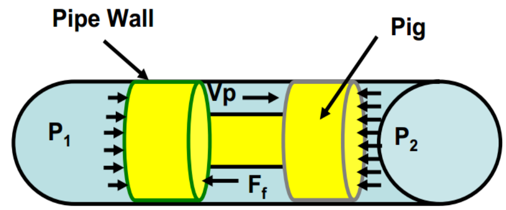

Figure 4: Flat disk model

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3. Material Modeling

3.1 Introduction

Polyurethane (PU) is the most commonly used material for both support and sealing cups of IPIG tools on account of their possessing the following properties are - Abrasion resistance, cut and tear resistance, higher load bearing ability, oil and petroleum resistance, microbial and corrosion resistance, non-marking, non-staining.

The PU grades are specified in terms of shore hardness values in terms of two prevalent scales namely Shore A or Shore D measured with a Durometer. The higher the durometer number the harder is the polyurethane. In general, polyurethanes typically ranging from 60 to 95 on the Shore A scale are used. The tear strength and elongation at break are other parameters by which the PU grades are specified. Tea strength is the tensile stress at the moment of rupture of the specimen and elongation at break is the elongation corresponding to tear strength. The two major grades of polyurethane material along with their properties considered for evaluation under this study are listed below in table 1.

Table 1: Properties of two grades of PU materials consid ered for study

Typical properties of compound no.	PO-650	PO-652
Shore Durometer (ASTM D2240-64T)	85A	95A
Yield Strength (ASTM D412-61T)	30MPa	35MPa
Tensile Modulus		
@50% Elongation	3.41MPa	7.49MPa

@100% Elongation	4.82MPa	10.71MPa
Tear Strength (KN/m)		
Split Tear(ASTM D470)	24.6	35.1
Compression-Deflection (Load Carrying Ability)		
5%	1.3MPa	3.2MPa
25%	6.8MPa	12.8MPa

3.2 Material Teasting of PO-650 & PO-652

Tensile tests were performed on samples obtained from one a cup made of PO-650 grade polyurethane material. Samples were cut from the cups into cylinder of about 12mm diameter using a die cutter. Tensile tests were carried out using an Instron 4505 universal test machine. Strains were determined using a Messphysik video extensometer capable of simultaneous measurement of axial extension. The stress -strain relationship for polyurethane material is highly non-linear and large strains values till failure are expected. The nominal stress and strain curve obtained is shown Fig 5, the material was seen to have a hardness value of 85 Shore.

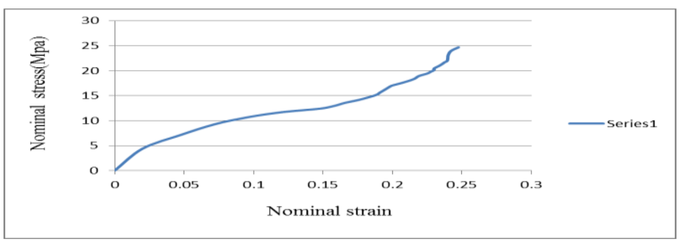

Figure 5: Nominal Stree vs. Nominal Strain

The stress-strain behavior of polyurethane is elastic (i.e. recoverable) but highly nonlinear. This type of material behavior is known as hyper elasticity. Hyper elastic constitutive laws are used to model materials that deform elastically when subjected to very large strains. They account for both nonlinear material behavior and large shape changes. Hyper elastic material is also Cauchy-elastic, which means that the stress is determined by the current state of deformation and not the path or history of deformation.

Unlike linear elastic material, in hyper elastic materials the stress-strain relationship derived from a strain energy density function is not a constant factor i.e. modulus of elasticity changes. A convenient way to define a hyper elastic material is to input the test data as such into the ABAQUS code. ABAQUS uses strain energy potentials to relate stresses to strains in hyper elastic materials. The different forms of strain energy potentials available are: a polynomial model (of which the Mooney-Rivlin is a particular case) and the Ogden model.

3.3 Mooney-Rivlin & Ogden Model

The Mooney-Rivlin case is obtained from the polynomial form of the hyper elastic model by setting the polynomial parameter, N, to one, i.e. the first order polynomial. The Mooney-Rivlin model

Table 2: Dimensional Text vs. Size

Sr. No	Dimension Text	Size
1	Nominal pipe diameter	304.8mm
2	Conical cup outer diameter	320 mm
3	Conical cup Inner diameter	150 mm
4	P.C.D of mounting bolts (Stainless steel spacer 12.5 I.D./15.5 O.D./ 18 mm long at all 16 places)	210 mm
5	Diameter of mounting bolts	M10
6	Number of bolt holes	16 Nos.
7	Thickness of cup at Inner diameter	25 mm

uses only linear functions of the deviatoric strain invariants (I1 , I2) and strain energy function (W) for this model is given by equation (4) :-

$$W = C_{01}\left(\bar{I}_2 - 3\right) + C_{10}\left(\bar{I}_1 - 3\right) + D_1\left(J - 1\right)^2 \quad \text{---------- (4)}$$

The initial shear modulus (G0) and the bulk modulus, K0, when N=1 are:

G0 = 2 (C10 + C01) K0 = 2/D1

Where C10, C01 and D1 are temperature-dependent material parameters. The D1 parameter allows for the inclusion of compressibility.

The Ogden strain energy potential is expressed in terms of the principle stretches. In ABAQUS, the form of the Ogden strain energy potential (w) is given by equation (5):-

$$W = \sum_{i=1}^{N} \frac{\mu_i}{\alpha_i}\left(\bar{\lambda}_1^{\alpha_i} + \bar{\lambda}_2^{\alpha_i} + \bar{\lambda}_3^{\alpha_i} - 3\right) + \sum_{i=1}^{N} \frac{1}{d_i}\left(J_{el} - 1\right)^{2i} \quad \text{-------- (5)}$$

Where λ1, λ2, λ3 are three principal stretch ratio which will provide a measure of the deformation. N is the order of the polynomial, μi, αi, and Di are temperature-dependent material properties. In this work, a third order polynomial form of the Ogden model was used.

4. Finite Element Model of Sealing Cup

4.1 Assumpation

The following assumptions have been made in the finite element modelling of the PU cups in determination of the stress values.

1. The centerline of IPIG matches with the centerline of the pipeline. So it has been assumed that the wear and friction are uniform all-round the circumference of seal. In real condition, due to weight of IPIG, friction was more at the bottom of cup which leads to uneven wear and leakage around the seal.

2. The force exerted on the pipe wall is mainly due to oversized cup and weight of the assembly. Therefore, most of the friction is due to the sliding contact between the seal and the pipe wall. This is almost always true when the pig is new. However as the seal wear out, the wall force and this assumption becomes less valid.

3. The cup has been assumed to contact the pipe wall all along its chamfered edge.

4. Non-linear properties have been considered for the polyurethane materials.

4.1 Geometery

The polyurethane sealing cup has been modeled using the modeling software available in ABAQUS. The relevant dimensions considered for the sealing cup are shown in Table-4.1a.

4.2 Load and Boundary Condition

Differential pressure (ΔP) is applied to the inner side of conical cup as shown in Fig 6. The average value of differential pressure generally experienced across the sealing cup is 0.1961Mpa. In Fig 7, Pipe normal wall force acting on cup on the outer periphery is shown.

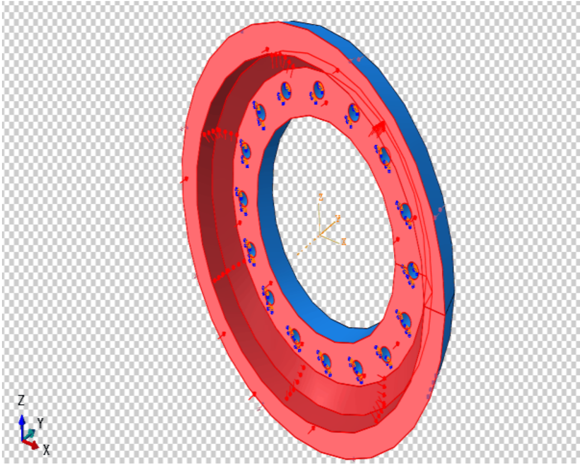

Figure 6: Differential pressure (ΔP) on Inner surface

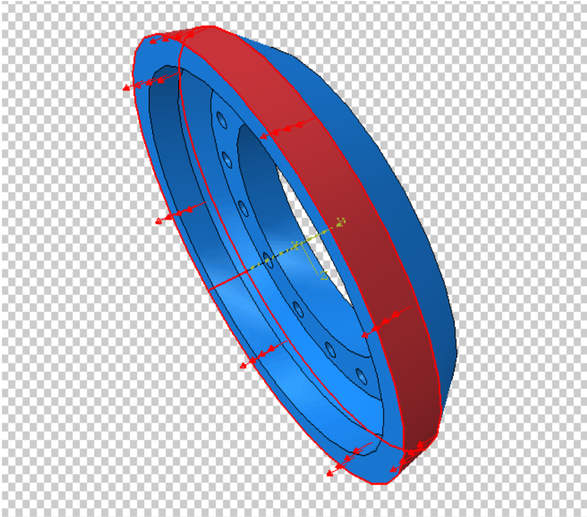

Figure 7: Wall force (W) on outer periphery

Friction force will act opposite to the motion of pig. It acts parallel to the outer periphery of cup as shown in Fig.8. The cup is mounted on pig by bolts. Therefore, the Encastre boundary condition has been applied on bolt holes which means completely fixed / clamped as shown in Fig. 9.

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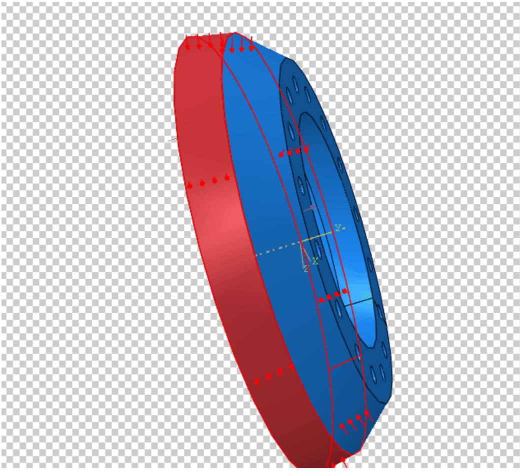

Figure 8: Friction force (F)

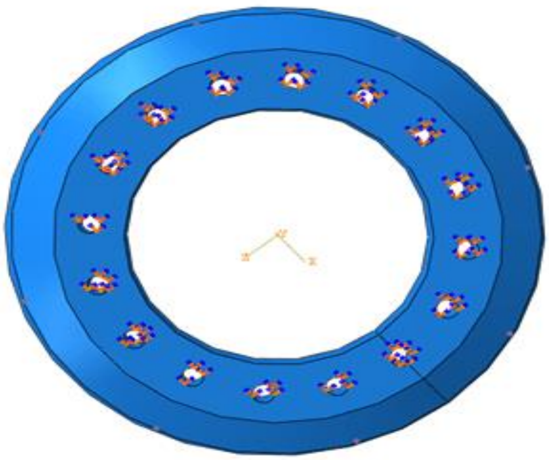

Figure 9: Encastre on bolt holes

5. Result and Discussion

5.1 FEM Result of Sealing Cup Made of PO-650

It is seen that for the differential pressure across the sealing cup ranges from 0.5Mpa to 1.9Mpa. When IPIG tool gets stuck inside the pipeline, a high pressure builds-up behind the cup which deforms the cup in opposite direction thereby over-turning the cup.

Table 3: FEM results for PO-650 material using Mooney-Rivlin model

Differential Pressure(ΔP) (MPa)	Wall Force, W (KN)	Friction force, F=0.3W (KN)	Von-mises Stress (MPa)	Max. deformation (mm)
0.5	20.2	6.0	2.64	1.28
1.0	39.8	11.9	5.37	2.57
1.5	54.6	16.3	8.25	2.02
1.9	76.7	23.4	9.98	6.33

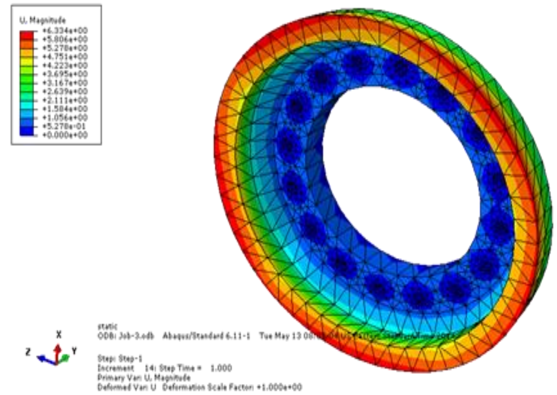

Figure 10: Deformation for Mooney-Rivlin model of PO-650 material cup at max (ΔP=1.9M

5.1 FEM Result of Sealing Cup Made of PO-652

The 12' (320mm) sealing cup design PU material PO-652 has also been modeled using the hyper-elastic model i.e. Mooney-Rivlin. The finite element analysis has been conducted using the ABAQUS code. Values of Von-mises stress and deformation values have been obtained for various differential pressure values as shown in table 4. The maximum stress induced and deformations for this material are seen to be less than that of PO-650.

Table 4: FEM results for PO-652 material using Mooney-Rivlin model

Differential Pressure(ΔP) (MPa)	Wall Force, W (KN)	Friction force, F=0.3W (KN)	Von-mises Stress (MPa)	Max. deformation (mm)
0.5	**20.2**	**6.0**	**2.34**	**0.62**
1	**39.8**	**11.9**	**5.37**	**2.57**
1.5	**54.6**	**16.3**	**8.12**	**2.72**
1.9	**76.7**	**23.4**	**9.47**	**2.98**

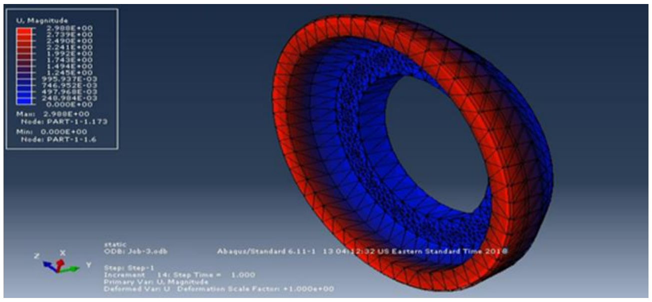

Figure 11: Deformation for Mooney-Rivlin model of PO-652 material cup at max (ΔP=1.9MPa)

6. Conclusion

1 Mooney-Rivlin model for hyper elastic materials is seen to best describe the chosen grades of polyurethane materials

2 The maximum stress and deformation values for the both the materials are seen to be within the safe range even at maximum differential pressure (1.8 MPa) when compared using the Von- Mises failure criteria.

3 The maximum stress concentration is seen to occur at the bolt holes which can initiate tearing of the cup.

4 The overturning pressure obtained through finite element analysis are seen to closely match the experimentally determined values

5 Between the two grades of polyurethane material, PO-652 is inferred to be better choice as the maximum stresses in the material are seen to be less as compared to the poly urethane material PO-650.

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