Part Number (Rev Level) and Part Name

KU661-SM19	Dilatant Failure

Basic Information

Engineer:	Samuel Hong, James Bevitt
Date:	7-Mar-22
Software:	Patran/Nastran 2018

Executive Summary

The objective of this project is to use the knowledge gained from ME661 and apply it to real life scenarios. In this case, FEA is used to analyze some part that attach to a pilot's helmet using Patran/Nastran 2018. The model was made by a KU team to help with Dilatant LLC, to make a design that will help reduce whiplash that may occur during flight.

Problem Statement

Two parts of the design were isolated and observed using Patran/Nastran2018. An analysis was required to see how the part's current geometry affects the stress concentrations in the part and possibly provide recommendations to changes in the geometry.

System Properties

Component	Weight (lbs)	Material
Cylinder	0.0028	Titanium
Total	0.0028	-

Component	Weight (lbs)	Material
Ball Joint	0.0107	Titanium
Total	0.0107	-

Material	E (ksi)	n	syield (ksi)	sultimate (ksi)	sallowable (ksi)	Density(lb/in3)
1018 Steel	1.68E+04	0.340	128.0	138	128	0.163

Model Geometry

Both models were made through SolidWorks2018. The cylinder, however, ran into some problems with it's original shape, so the model was then made through Patran/Nastran2018 by making a surface made with seven points, connected them with curves, then made into a surface by using the vertices option. It then had the mesh sweeped 360° to make the cylinder shape.

Mesh (figures located in Mesh worksheet)

The Cylinder was a troublesome model to deal with. The surface was made with a Quad4 with an AutoMesh Paver. It was then sweeped 360° to form a cylinder. Elements 1-304 forms the surface which has the Quad4 mesh. From 305-18544, the Elements have a Hex8 mesh.

The ball joint was created after applying a significant number of mesh seeds to the part. The hole and curved edge connecting to the sphere used 32 element mesh seeds, the short edgeconnecting to the sphere and long plate edges used 16 element mesh seeds. The outermost face opposite of the sphere has an 8 element mesh seed on the short edge and 16 element mesh seed on the long edges. After creating the mesh seeds, a Tetrahedral mesh was applied to the solid.

Case	*Element Type*	*AutoMesh*	*# of Elements*	*Total Elements*	*Nodes*	*Dof*
Cylinder1	*Quad4/Hex8*	*Paver*	*40*	*4223*	*5680*	*17040*
Cylinder2	*Quad8/Hex20*	*Paver*	*30*	*510*	*3720*	*11160*
Cylinder3	*Quad8/Hex20*	*Paver*	*20*	*340*	*2480*	*7440*
Cylinder4	*Quad8/Hex20*	*Paver*	*10*	*187*	*1240*	*3720*
Ball Joint 1	*Tet4*	*Isomesh*	*8228*	*8228*	*1871*	*5613*
Ball Joint 2	*Tet10*	*Isomesh*	*8228*	*8228*	*12854*	*38562*
Ball Joint						
Ball Joint						

Loads and Restraints (mesh with boundary conditions shown in workshets LR1, LR2, etc.)

Load Case 1: Cylinder

Load Type	Location	Magnitude	Direction	Notes
Force	Inner Lip Top Face	1	Positive Y	Force applied on the edge of lip

Restraint Case 1: Cylinder

Restraints	Location	Magnitude	Direction	Notes
Displacement	Bottom Face	0	All	

Load Case 2: Ball Joint

Load Type	Location	Magnitude	Direction	Notes
Force	Inside Right Hole Node	1	Positive Z	

Restraint Case 2: Ball Joint

Restraints	Location	Magnitude	Direction	Notes
Displacement	Sphere	<0,0,0>	All	Displacement conditions in rough circular shape where cylinder contacts.
MPCs	Right Half of Hole Nodes	Uz=Uz	N/A	Deforms hole evenly in Z-direction

Analysis of Results (plots shown in worksheets Results 1, Results 2, etc.)

The effective stress concentration factor was determined by comparing the maximum stress to a reference stress, which is the stress the part would have had if the part had an axial load applied evenly over its smallest cross-sectional area. The reference area for the cylinder was taken at its base, with a cross-sectional area measured to be .043 sq in. The reference area for the ball-socket joint was taken at the thinnest cross-section of the beam where the hole is, determined to be .0116 sq in.

A quirk in the FEA program when dealing with total load resulted in much larger loads being applied than the intended unit loads. For the purpose of this lab, which is to estimate stress concentration factors, the reference stress is adjusted to the actual loads that were applied in the OLOAD section, so this should not significantly effect the resulting concentration factors.

Load Case	Max Stress	Reference Stress	Effective Stress Concentration Factor
Cylinder 1	12000.0000	920	13
Cylinder 2	18400.0000	690	27
Cylinder 3	13900.0000	460	30
Cylinder 4	8120.0000	460	18
Ball Joint 1	102.0000	86.2	1.18
Ball Joint 2	387.0000	86.2	4.49

Other Areas of Note						
Load Case	Notes					
Cylinder						
Ball Joint						

Model Verification Checks

Case1: Cylinder1

Total Applied Loading vs Resultant

Load Type	Fx	Fy	Fz	Mx	My	Mz
OLOAD	0.00E+00	4.00E+01	0.00E+00	-2.24E-08	0.00E+00	9.31E-09
SPCFORCE RESULTANT	3.75E-14	-4.00E+01	-1.80E-15	2.24E-08	1.59E-15	-9.31E-09

Epsilon vs External Work

Epsilon	-3.09E-15
External Work	1.62E-03
Residual Work	-4.99E-18

Case2: Cylinder2

Total Applied Loading vs Resultant

Load Type	Fx	Fy	Fz	Mx	My	Mz
OLOAD	0.00E+00	3.00E+01	0.00E+00	1.27E-07	0.00E+00	6.89E-08
SPCFORCE RESULTANT	1.77E-13	-3.00E+01	-2.60E-14	-1.27E-07	-5.17E-15	-6.89E-08

Epsilon vs External Work

Epsilon	4.97E-15
External Work	1.00E-03
Residual Work	4.98E-18

Case3: Cylinder3

Total Applied Loading vs Resultant

Load Type	Fx	Fy	Fz	Mx	My	Mz
OLOAD	0.00E+00	2.00E+01	0.00E+00	3.35E-08	0.00E+00	7.45E-09
SPCFORCE RESULTANT	-5.92E-14	-2.00E+01	1.45E-13	-3.35E-08	-1.44E-14	-7.45E-09

Epsilon vs External Work

Epsilon	-7.45E-09
External Work	4.82E-04
Residual Work	-3.59E-12

Case4: Cylinder4

Total Applied Loading vs Resultant

Load Type	Fx	Fy	Fz	Mx	My	Mz
OLOAD	0.00E+00	2.00E+01	0.00E+00	7.82E-08	0.00E+00	7.83E-09
SPCFORCE RESULTANT	-3.55E-14	-2.00E+01	-7.86E-14	-7.82E-08	-2.37E-14	-7.83E-09

Epsilon vs External Work

Epsilon	7.42E-15

External Work	4.18E-04
Residual Work	3.10E-18

Model Verification Checks

Case1: Ball Joint (Tet4)

Total Applied Loading vs Resultant

Load Type	Fx	Fy	Fz	Mx	My	Mz
OLOAD	0.00E+00	0.00E+00	1.00E+00	4.87E-02	5.80E-04	0.00E+00
SPCFORCE RESULTANT	-2.44E-15	7.71E-16	-1.00E+00	-4.86E-02	-5.80E-04	4.39E-16

Epsilon vs External Work

Epsilon	2.16E-14
External Work	4.95E-07
Residual Work	1.07E-20

Case2: Ball Joint (Tet10)

Total Applied Loading vs Resultant

Load Type	Fx	Fy	Fz	Mx	My	Mz
OLOAD	0.00E+00	0.00E+00	1.00E+00	4.87E+02	5.80E-04	0.00E+00
SPCFORCE RESULTANT	-4.69E-15	3.94E-16	-1.00E+00	-4.87E-02	-5.11E-04	-4.18E-16

Epsilon vs External Work

Epsilon	-7.85E-14

	External Work	7.25E-07						
	Residual Work	-5.68E-20						

Conclusions/Recommendations

Under such loads, the cylinder would definitely not be able to withstand a impulse force of 15 G, judging by the fact that under a total point load of 40 lbs, the material yielded. The main reason could perhaps because of the thickness of the lip was not thick enough and not enough support around the edges. Different alloys could also perhaps strengthen the model. As for the ball joint, it withstood better than the cylinder, but at the end, it will not be enough to withstand a 15G impulse force.

Both models were terribly small and the group did predict that they would not be able to resist the force well enough. However, it did show how the models deformed and where improvements can be made.

Both the cylinder and the ball joint results had results with a somewhat mediocre results. However, not enough mesh analysis were made for the ball joint to see any convergence, and the cylinder's data had a big range because of the inconsistency of the loading.

The loading was down on each FEM node on the model in Patran/Nastran2018, which resulted in the data showing the y load being larger than 1. Each of the Y loads are the sum of all the node loads, so in a 30 element mesh, a total of 30 lbf was applied.

For the ball joint, the geometry was very difficult to handle, as well as the meshing on the spherical part.

There are definitely recommendations for future cases like these models. Having the same number of elements and changing the topology wouuld have brought out a more consistent data, and perhaps see the convergence. For the ball joint, saving multiple files with a basic model that will have the same loading but different mesh will save a lot more time and more efficient.