ABCO AUTOMATION

CASE STUDY

STRESS & DEFLECTION ANALYSIS OF AN EXISTING ROBOT MOUNT

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OF AN EXISTING ROBOT MOUNT

INTRODUCTION

Study on an existing robot mount - stress and deflection analysis - to determine if a robot mounted horizontally on a stand would fail or deflect excessively.

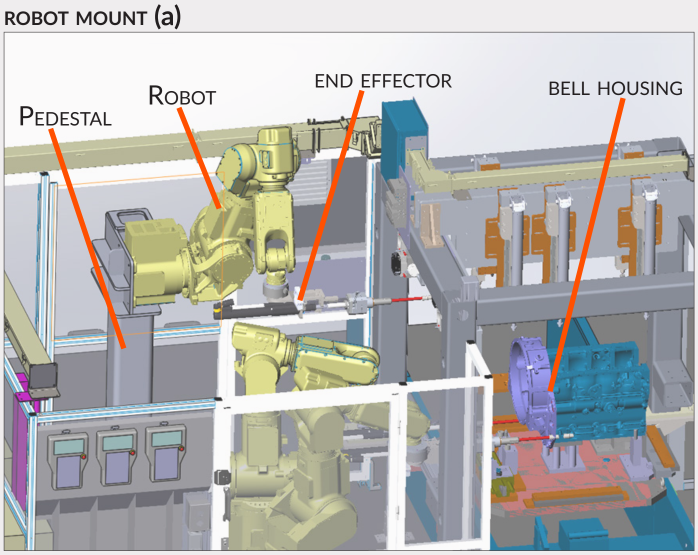
PROBLEM STATEMENT

- What is the stress and deflection?
- Robot weight: 510 pounds
- End effector weight: 41 pounds
- Moment loads at mounting plate
  X direction and Z direction
- Weight adds shear at holes
- Main concern is pullout of bolts
- Deflection of end effector at bell housing

ANALYSIS

An overall orientation picture (a) showsthe cell and the relationship to the different parts. We see in this image the breakout of the parts of concern, making the problem statement more clear.

ROBOT MOUNT (a)



The problem statement lays out the main factors and concerns. As a side note, bell housing is another name for flywheel housing. On older engines, this part looked more like a bell; on the old Model T this part was called a hog's head due to its shape and resemblance of the head of a hog.

The Mass Properties tool was used to calculate the weights and moment arms (b).

LOADS USING MASS PROPERTIES TOOL (b)

Inputs				
	Weight of Robot	510	lbs	
	Lever arm of Robot	24	inches	X
	Lever arm of Robot	3	inches	Z
	Weight of End Effector	41	lbs	
	Lever arm of End Effector	18	inches	Z
	Lever arm of End Effector	24	inches	X
	Distance between bolts	226	mm	X and Z
	Number of bolts	4		
	Size of bolts	0.625	inches	diameter, treaded into plate
Outputs				
	Moment of Robot	12,240	in-lbs	X
	Moment of Robot	1,530	in-lbs	Z
	Moment of End Effector	984	in-lbs	X
	Moment of End Effector	738	in-lbs	Z
	Total X moment	13,224	in-lbs	X
	Distance between bolts	8.898	inches	
	Pullout force top bolts	743	lbs	per hole
	Force at bottom	1,486	lbs	Spread over mount
	Shear at bolts	127.4	lbs	per hole
	Down force at right bolts	6.1	lbs	per hole
	Down force at left bolts	260.9	lbs	per hole

The shear forces are assumed to be resisted by the bolts alone, when in fact, some of the load would be taken up by friction between the two parts. The forces were calculated using standard physics and geometry. Calculations should be double checked by another person, just like drawings.

Next, the deflection was calculated (c). A new model was produced; it is one part and not a welded assembly. The post is an extrusion and not a beam. Fillets were added to represent the welds. The base of the robot adds to the rigidity of the mount.

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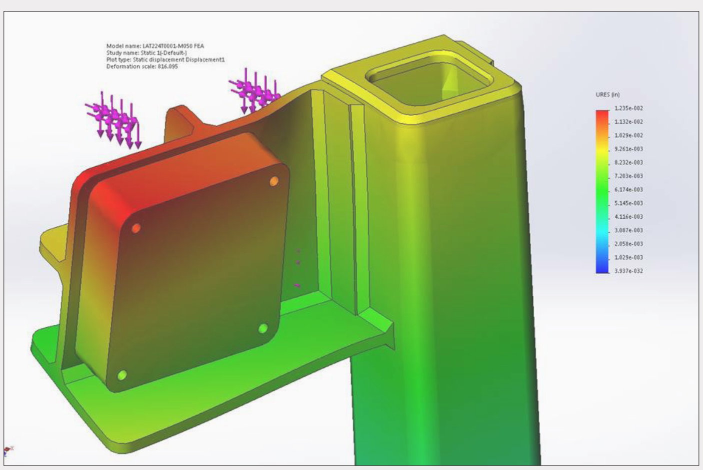
This was modeled by adding an extrusion to represent the base. The thicness is not important as long as it is thick enough to ensure this area acts in unison. To be 100% accurate, the robot base could be added to the part as a series of extrusions and cuts.

Assemblies are also possible but are harder to work with and the parts of the assembly usually have to be remodeled anyway. Any holes or small features that are not highly stressed or that do not contribute to stress or strain can be removed or left off the new model. Exceptions are holes in stressed areas that act as stress risers.

The calculated forces at the bolt holes were applied. This is represented by the purple arrows shown in the image (c). Some of the arrows are obscured by the exaggerated deflection of the mount.

DEFLECTION (c)



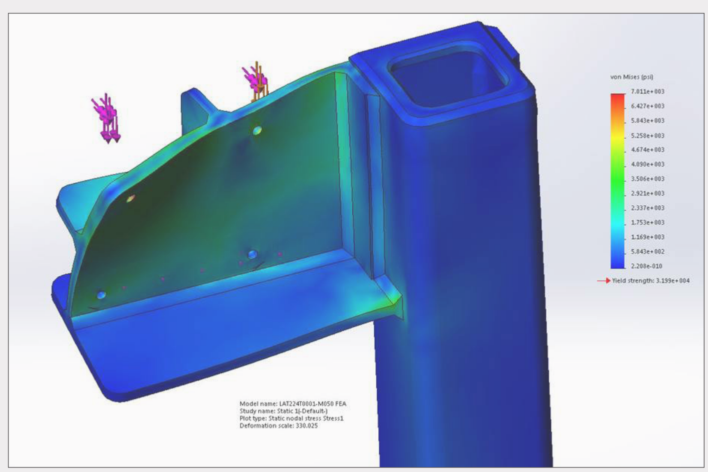
Base of robot adds stiffness at mount, modeled using thickened plate. Slope at mount is 0.021°, which translates to 0.030 inch downward deflection at bell housing.

The slope at the base was extended to the socket at the bell housing. The downward deflection can be compensated for in the programming of the robot.

The stress is calculated and the thickened plate representing the robot base has been suppressed.

FEA results must be interpreted, especially at small features such as the bolt holes due to the size of the elements in the finite element analysis. Finer elements will give greater accuracy but increase computation time and computer memory use. In this case the maximum indicated stress is about 7,000 psi, but the indicated stress in the surrounding area is about half this value. The strength at the bolt hole was checked by calculating the maximum pullout force of the threads. The results are nearly the same as the average maximum stress of 3,500 psi would indicate (d).

DEFLECTION (d)



The indicated maximum stress from FEA is probably high.
Average maximum stress is about 3,500 psi.
Allowable stress is about 32,000 psi for mild steel.

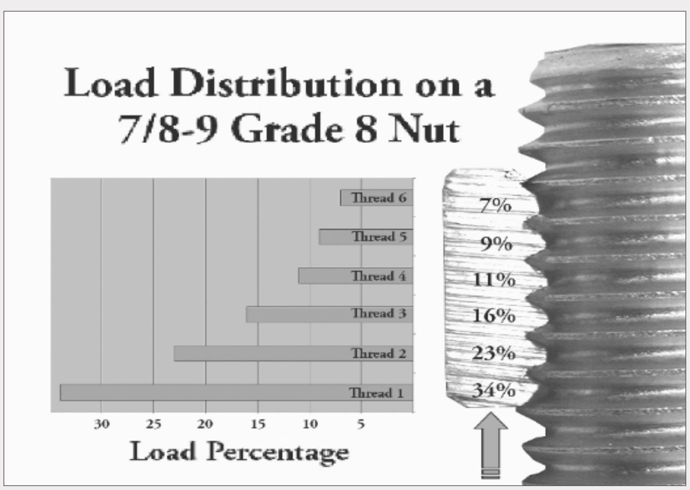
PULLOUT CHECK
Pullout force at upper bolts is 738 pounds
Bolt is 5/8- 11, tapped into plate
Tensile stress at major dia. of bolt= 2,400 psi
Grade 5 bolt is good for about 110,000 psi
The tread in the plate will strip first at about 5,800 lbs

The stress in the highest loaded bolt was first checked by a simple force/area calculation. Grade 5 bolts were assumed and should be used.

Next the pullout of the threads was checked. The next slide shows how this is done. The pullout at the treads in the plate is 5,800 pounds while the force is 738 pounds. The threads in the plate are weaker than the bolt. This gives a factor of safety of about 8:1.

We referred to a case study conducted by Fastenal® on threads in a nut, but this applies to threads in plates, as well (e).

THREAD LOAD DISTRIBUTION* (e)



The first thread takes 34% of the load.
It fails first, putting 34% on the second thread which then fails. The third thread follows and so on until the bolt pulls out.

* Source: www.fastenal.com/en/78/screw-thread-design

Softer material or different size fasteners will give slightly different results, but the general principles are the same.

The first thread is more stressed than the remaining threads by about 234%. This means that the nut would start to strip at about 42% of the indicated pullout force.

The force is basic shear stress times area, but the first thread takes 34% of the force (f). It will fail first. The other threads contribute to part of the force. The conservative approach is to use the 34% figure. The actual force at failure is about 42% of the calculated force to pull out the bolt. In this case, the calculated force to pull out is 17,131 pounds, so the bolt would be expected to fail at about 7,200 pounds. 34% gives 5,824 pounds. The actual fraction that the first thread takes depends on many factors, such as the quality of the threads, the yield point of the nut, and the size of the fastener.

INTERNAL PULLOUT STRENGTH* (f)

Thread Pullout					
	Inputs				
		Threads per inch	11		
		Length of thread	0.625	Inch	
		Effective length of threads	0.636364	Inch	7 threads
		Ds min	0.6234	Inch	Minimum major diameter of external threads
		En max	0.5644	Inch	Maximum pitch diameter of internal threads
		Shear strength of plate	16,000	psi	
	Outputs				
		Area of thread	1.071	sq inches	
		Force to pull out	17,131	lbs	All threads
		Force at first thread	5,824	lbs	First thread takes 34% of the load
	Source: https://www.fastenal.com/en/78/screw-thread-design				

The thread strength is shear strength X area, first thread takes 34%

* Source: www.fastenal.com/en/78/screw-thread-design

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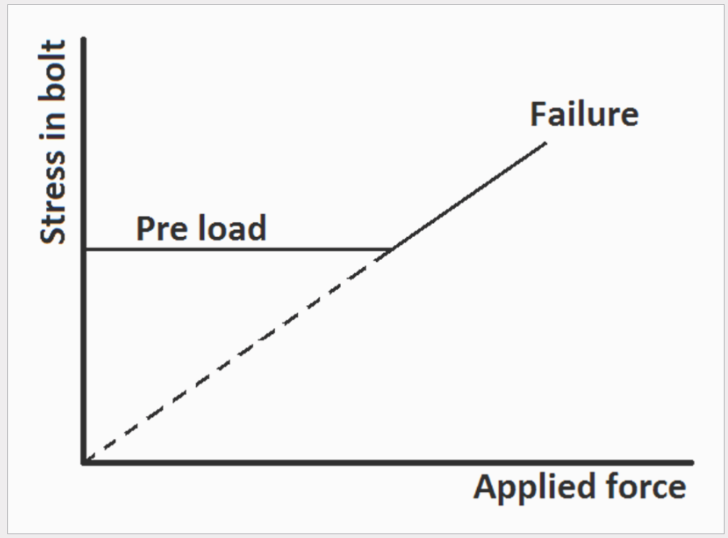
Bolts are preloaded by the torque applied during assembly. If the preload is more than the strength of the bolt, it will fail when the torque is applied, such is the case when bolts are over-torqued. If properly preloaded, the stress in the bolt will remain constant until the applied force is more than the preload. After that point, the stress will increase until failure (g).

EFFECT OF PRE LOAD ON BOLT (g)



Bolt is stressed at pre load level until applied stress becomes dominate.

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