Jon Berkowitz ______________________
Thomas Dwyer ______________________
James McClay ______________________
Daniel Reader ______________________
Brad Sims ______________________

FORMULA SAE STEERING SYSTEM

SPONSORED BY: UCD Formula SAE

FACULTY MENTOR: Professor Margolis

FINAL REPORT

EME-185A/B
PROFESSOR C. DAVIS
TEACHING ASSISTANT E. JORDAN

June 3, 2006

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ABSTRACT / EXECUTIVE SUMMARY

The UC Davis Formula SAE Team has designed a car to compete in the 2006 Formula SAE West competition on June 14th. In past years, the team has implemented two different types of steering systems, both with little success. In 2002, an off-the-shelf steering rack was purchased, but due to the unique requirements of FSAE vehicles, compromises were made which cause poor steering geometry and compliance in the system. Additionally, this unit was very expensive. In 2004, the team manufactured its own steering rack, but the design had many flaws. The rack housing was comprised of 5 pieces bonded together and the rack gear was a three piece bolted assembly. The intent was to reduce cost and increase manufacturability, but the result was a steering system with terrible binding problems due to excessive deflections and unnecessary weight and complexity.

The goal for this project was to design a steering system for the 2006 FSAE car to address all the shortcomings of previous designs, yet retain the low cost, manufacturability, and low weight. This steering system had to also comply with all aspects section 3.2.4 of the 2006 Formula SAE Rulebook. A multiple piece steering rack housing design was chosen, but unlike the 2004 design, minimizing compliance was emphasized. The base of the design is an aluminum tube running the entire length of the housing which supports both ends of the rack gear through bronze bushings. A carefully shaped notch in the tube exposes the rack gear teeth and precisely locates the pinion gear housing such that the gears interface properly. The pinion gear housing is firmly fixed to the tube through a clamping mechanism which eliminates any possible movement through the geometry of the mating surfaces. A dual pinion gear design allows for improved ergonomics and a straight steering shaft which does not require any flexible joints. The dual pinions allow the pinion housing to be located above the rack centerline while maintaining steering wheel motion that correlates with the correct steering direction. Another key element of the steering rack design is the custom round rack gear. High costs and limited selection of round rack gears inspired the creation of a custom rack gear from 1042 TGP steel bar using a wire EDM process in the student machine shop.

Upon final assembly of the completed system, the steering rack was found to perform flawlessly. There is very little backlash between the gears and operation of the system is quite smooth. The final weight of the steering rack, including mounts is 4.25 lbs. This is within 3% of the predicted weight according to the final 3D CAD model of the system. The new design has increased leg clearance for the driver, decreased weight by 2.25lbs, and reduced backlash in the system considerably. The full vehicle is still being assembled so complete testing has yet to be performed. Once the vehicle is complete, brief testing will be complete prior to competition with more extensive testing planned after competition for future design improvements.

FINAL LAYOUT DRAWINGS OF SYSTEM AND SUBSYSTEMS

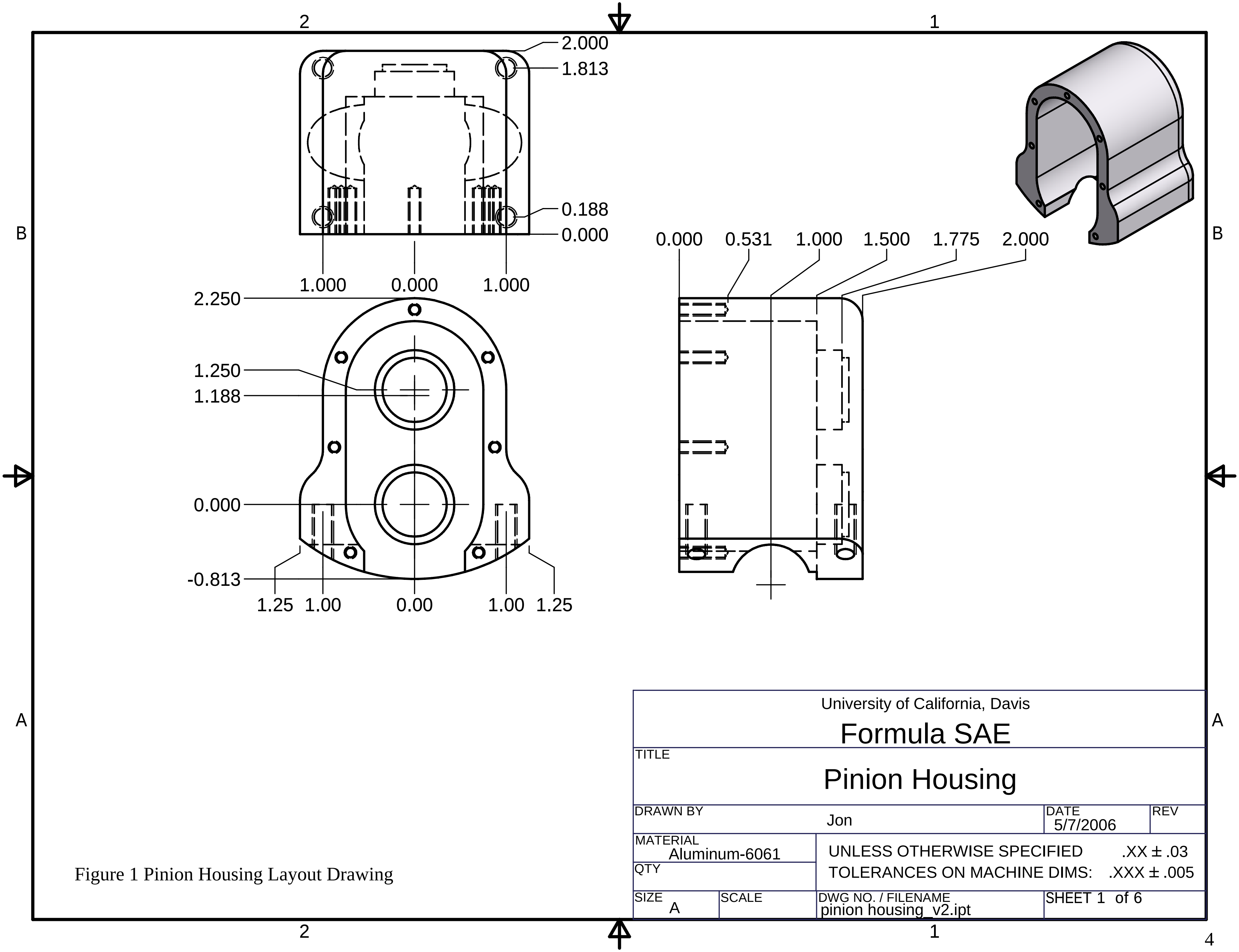
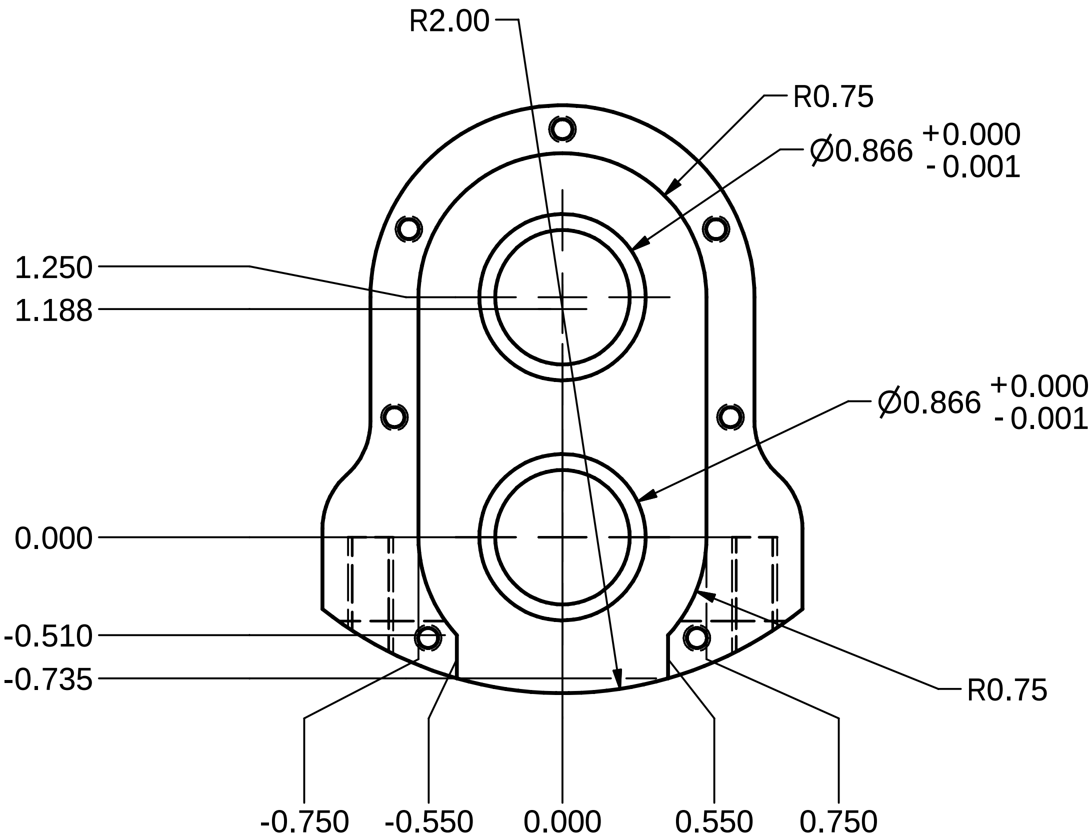
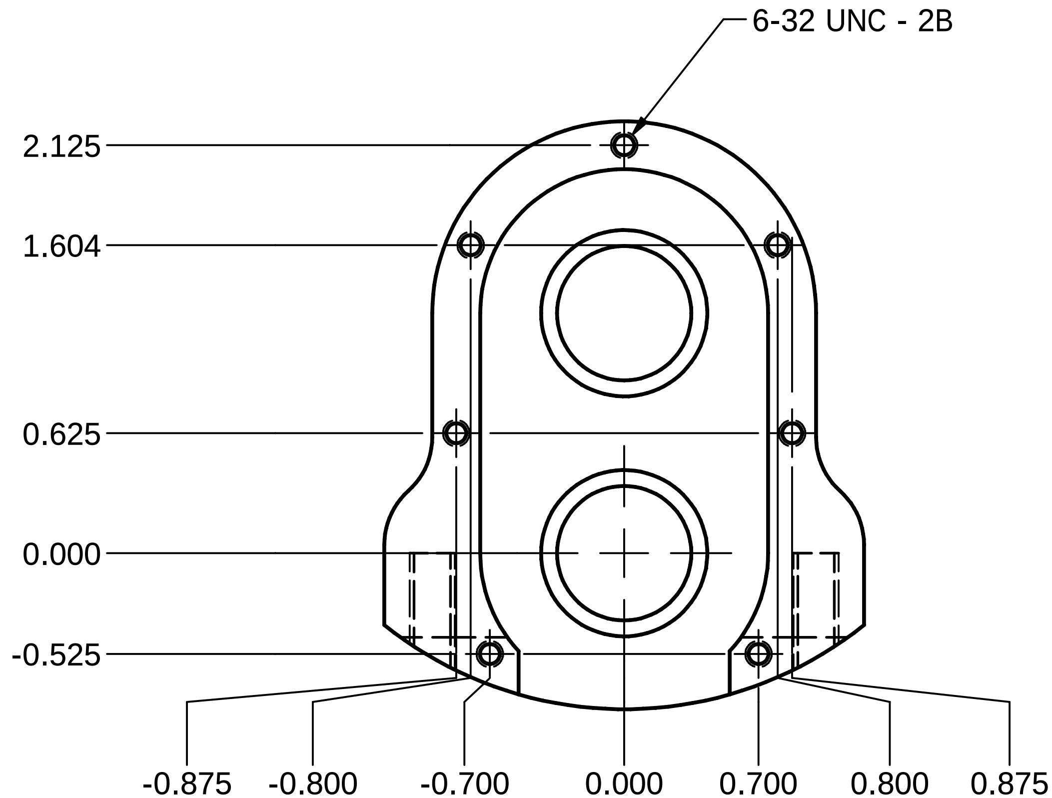


Figure 1 Pinion Housing Layout Drawing



University of California, Davis Formula SAE				
TITLE Pinion Housing				
DRAWN BY Jon			DATE 5/7/2006	REV
MATERIAL Aluminum-6061		UNLESS OTHERWISE SPECIFIED .XX ± .03 TOLERANCES ON MACHINE DIMS: .XXX ± .005		
QTY				
SIZE A	SCALE	DWG NO. / FILENAME pinion housing_v2.ipt	SHEET 2 of 6	

Figure 2 Pinion Housing Layout Drawing



University of California, Davis

Formula SAE

TITLE

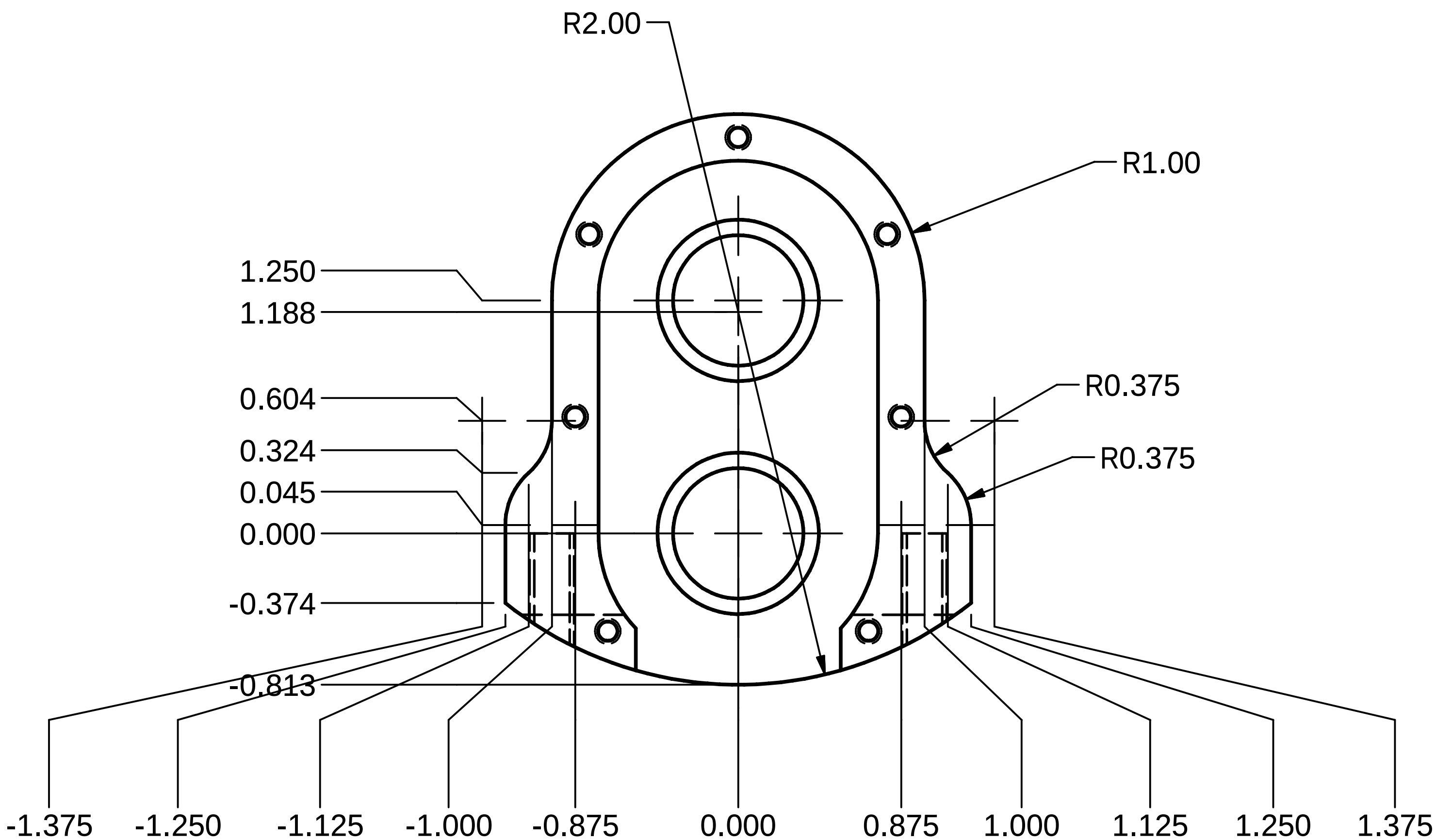
Pinion Housing

DRAWN BY	DATE	REV
Jon	5/7/2006	

MATERIAL Aluminum-6061	UNLESS OTHERWISE SPECIFIED .XX ± .03
QTY	TOLERANCES ON MACHINE DIMS: .XXX ± .005

SIZE	SCALE	DWG NO. / FILENAME	SHEET
A		pinion housing_v2.ipt	SHEET 3 of 6

Figure 3 Pinion Housing Layout Drawing



University of California, Davis Formula SAE			
TITLE Pinion Housing			
DRAWN BY Jon		DATE 5/7/2006	REV
MATERIAL Aluminum-6061	UNLESS OTHERWISE SPECIFIED .XX ± .03 TOLERANCES ON MACHINE DIMS: .XXX ± .005		
QTY			
SIZE A	SCALE	DWG NO. / FILENAME pinion housing_v2.ipt	SHEET 4 of 6

Figure 4 Pinion Housing Layout Drawing

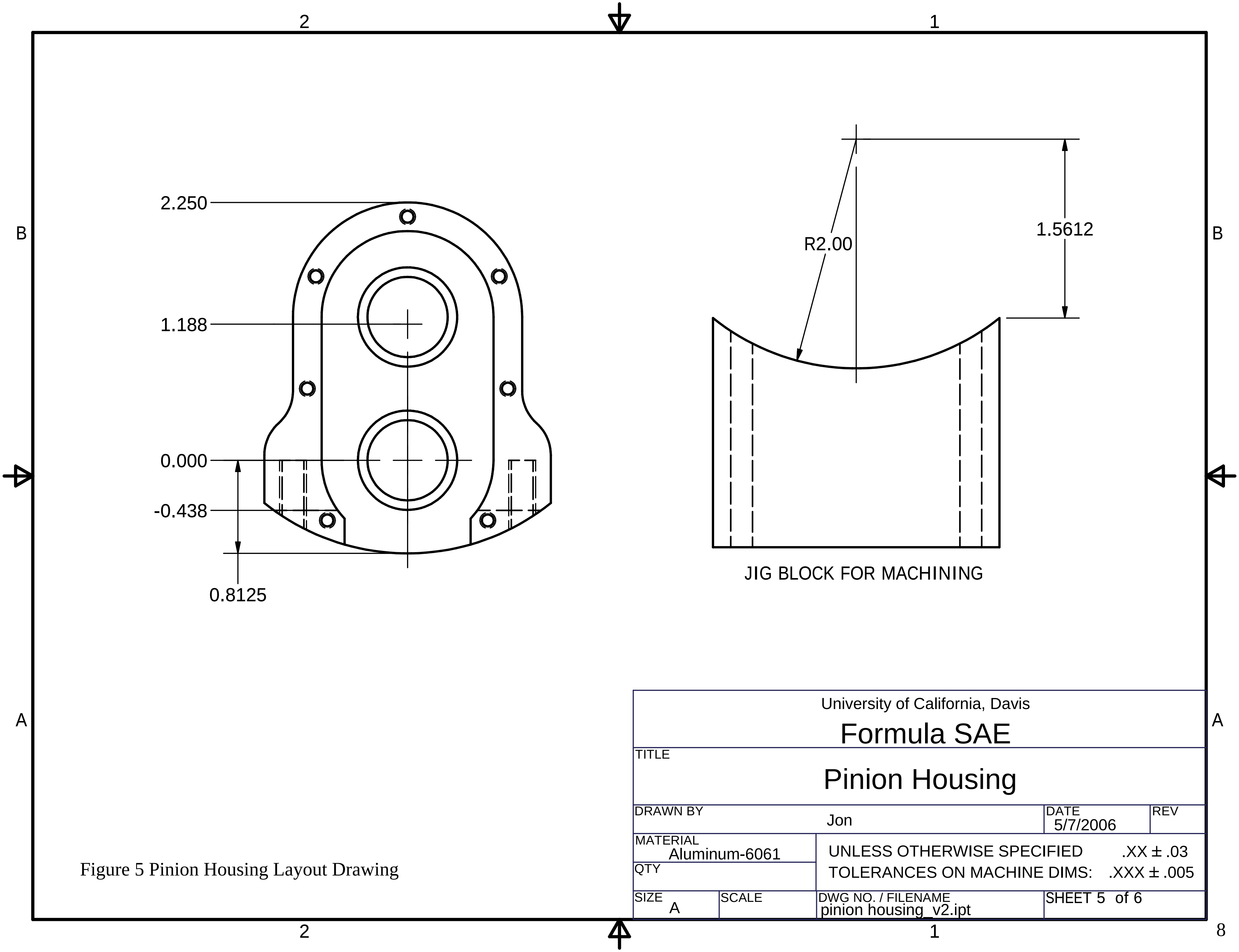


Figure 5 Pinion Housing Layout Drawing

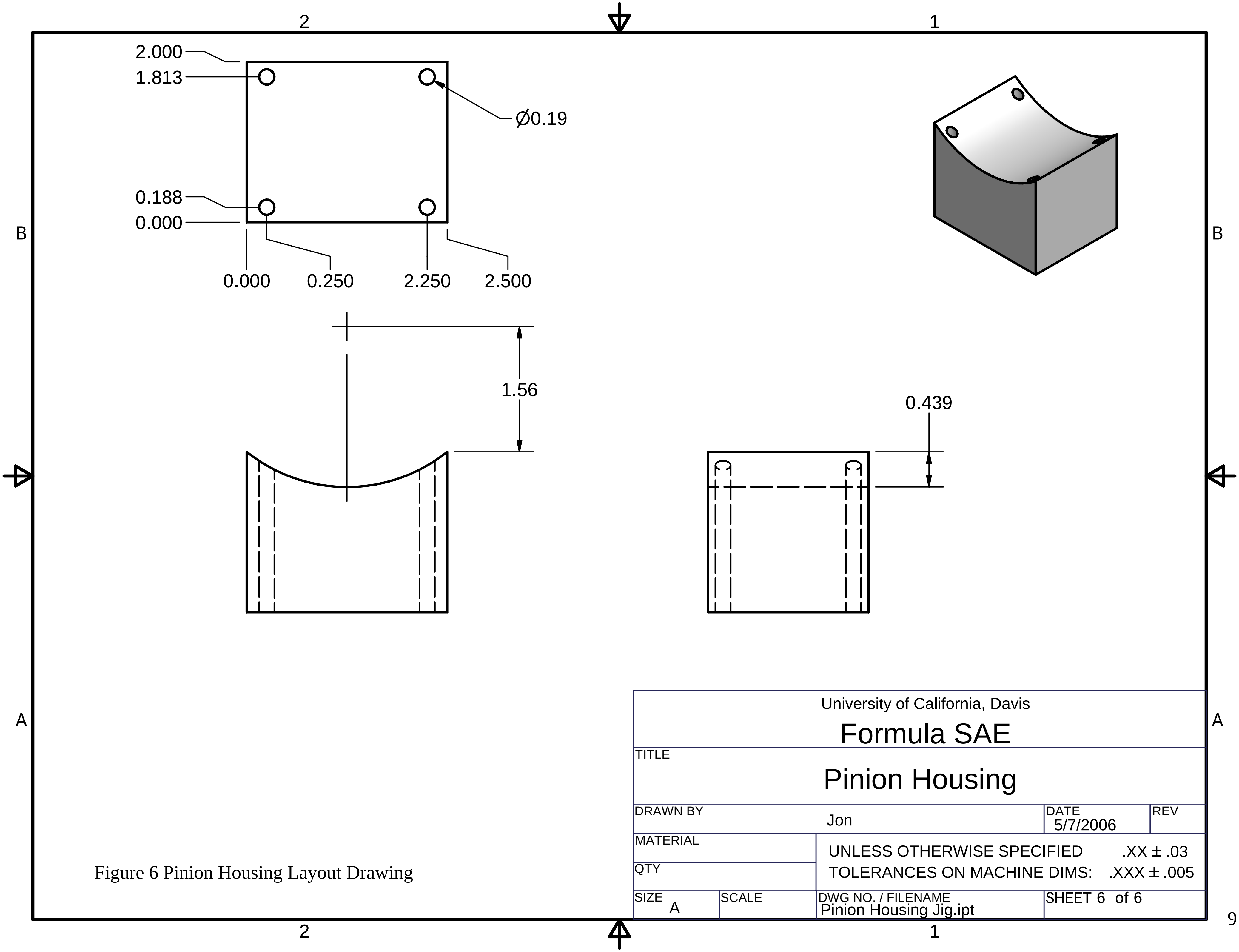
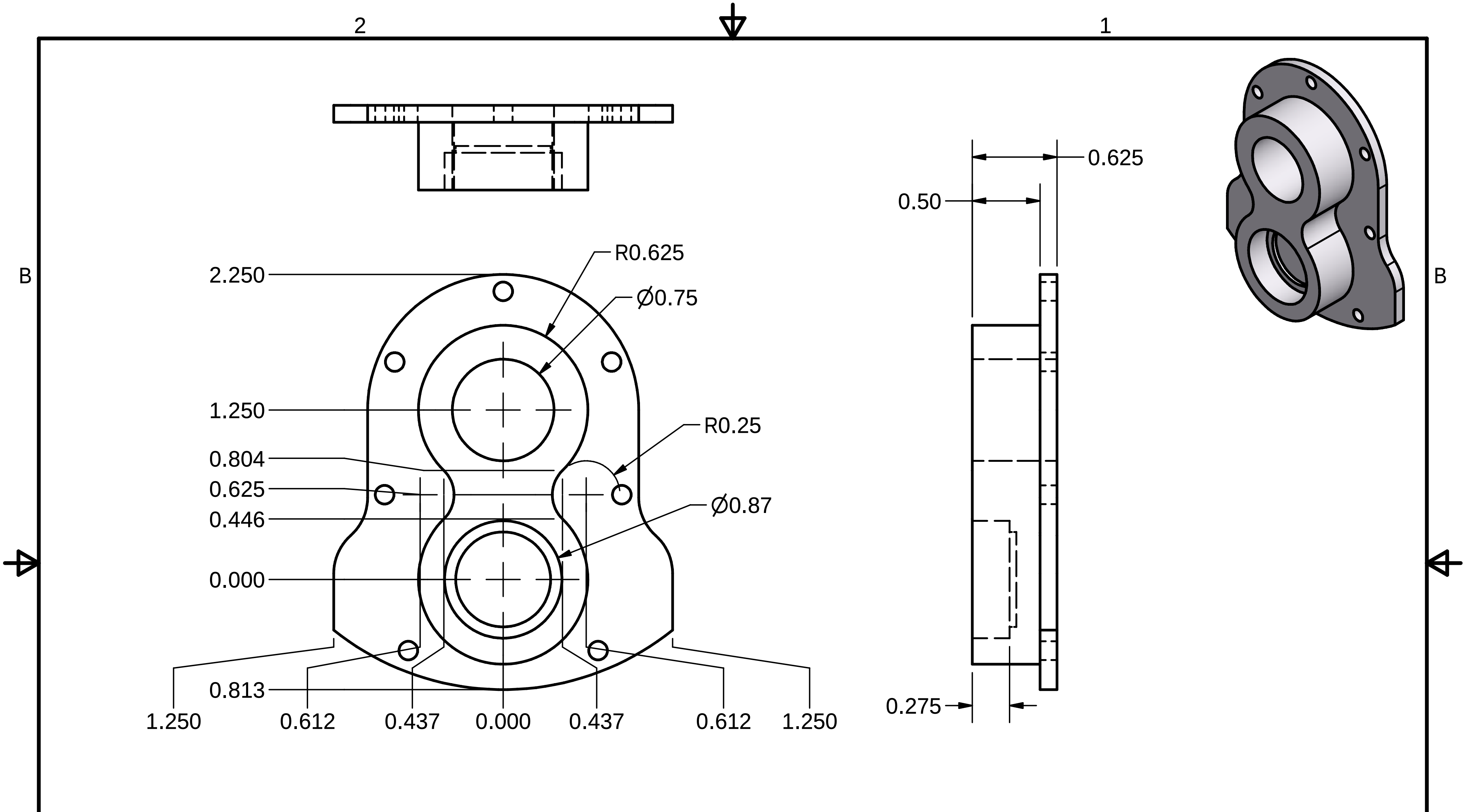


Figure 6 Pinion Housing Layout Drawing



University of California, Davis Formula SAE			
TITLE Pinion Housing Cap			
DRAWN BY Jon		DATE 5/7/2006	REV
MATERIAL Aluminum-6061	UNLESS OTHERWISE SPECIFIED .XX ± .03 TOLERANCES ON MACHINE DIMS: .XXX ± .005		
QTY			
SIZE A	SCALE	DWG NO. / FILENAME pinion housing cap_v2.ipt	SHEET 1 of 3

Figure 7 Pinion Housing Cap Layout Drawing

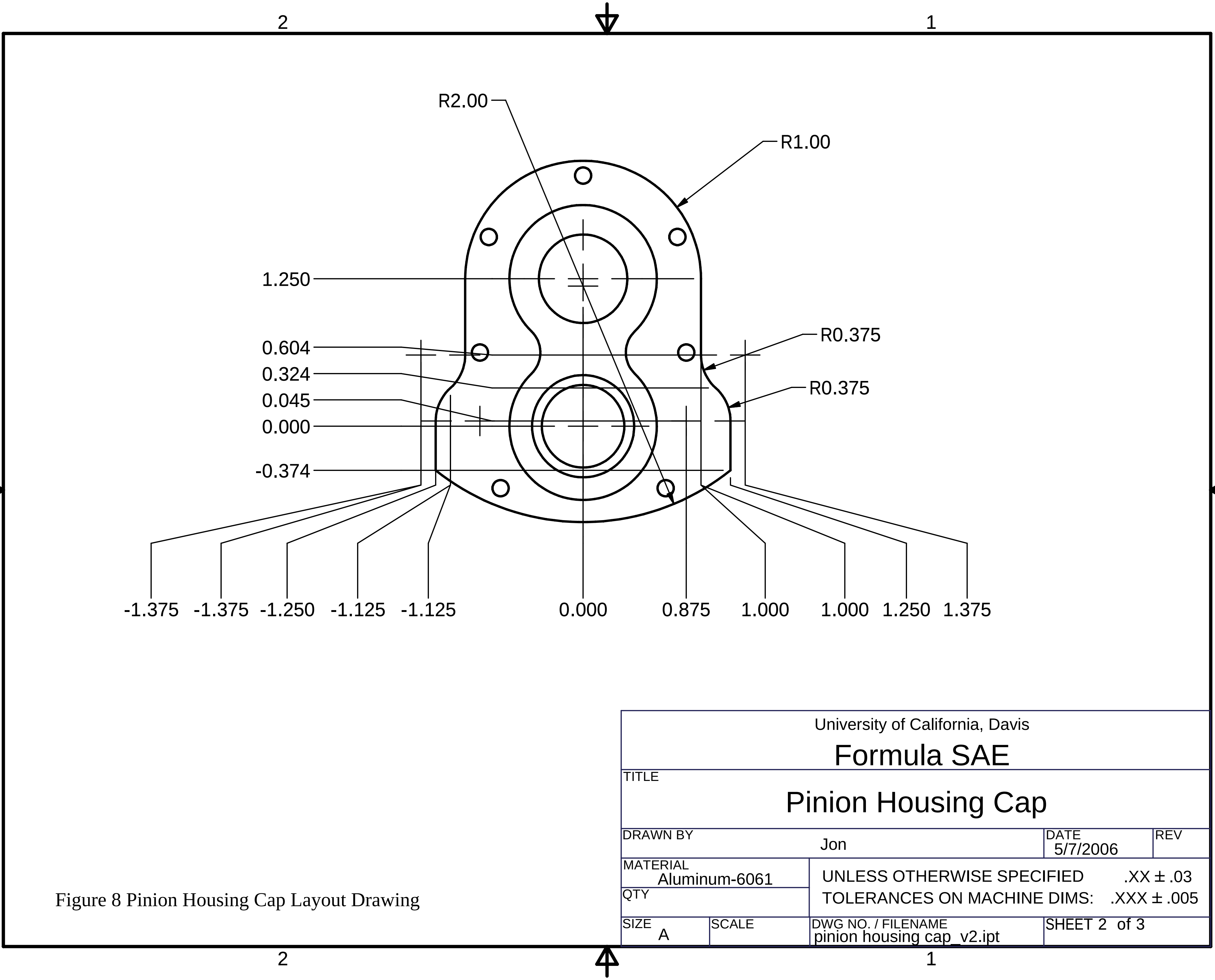
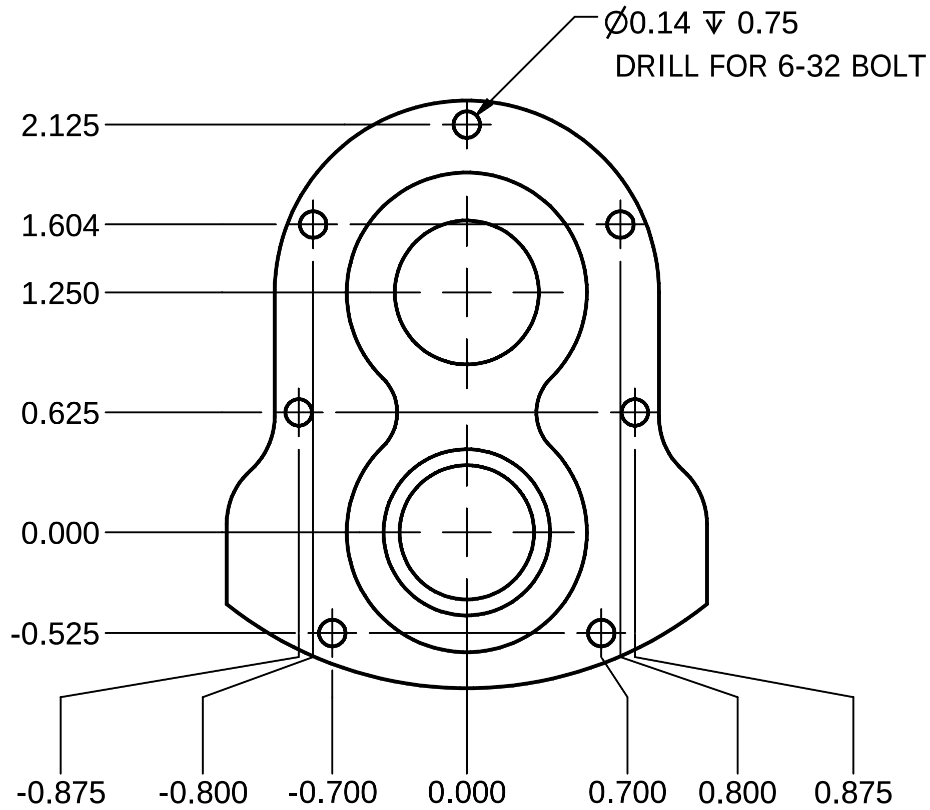
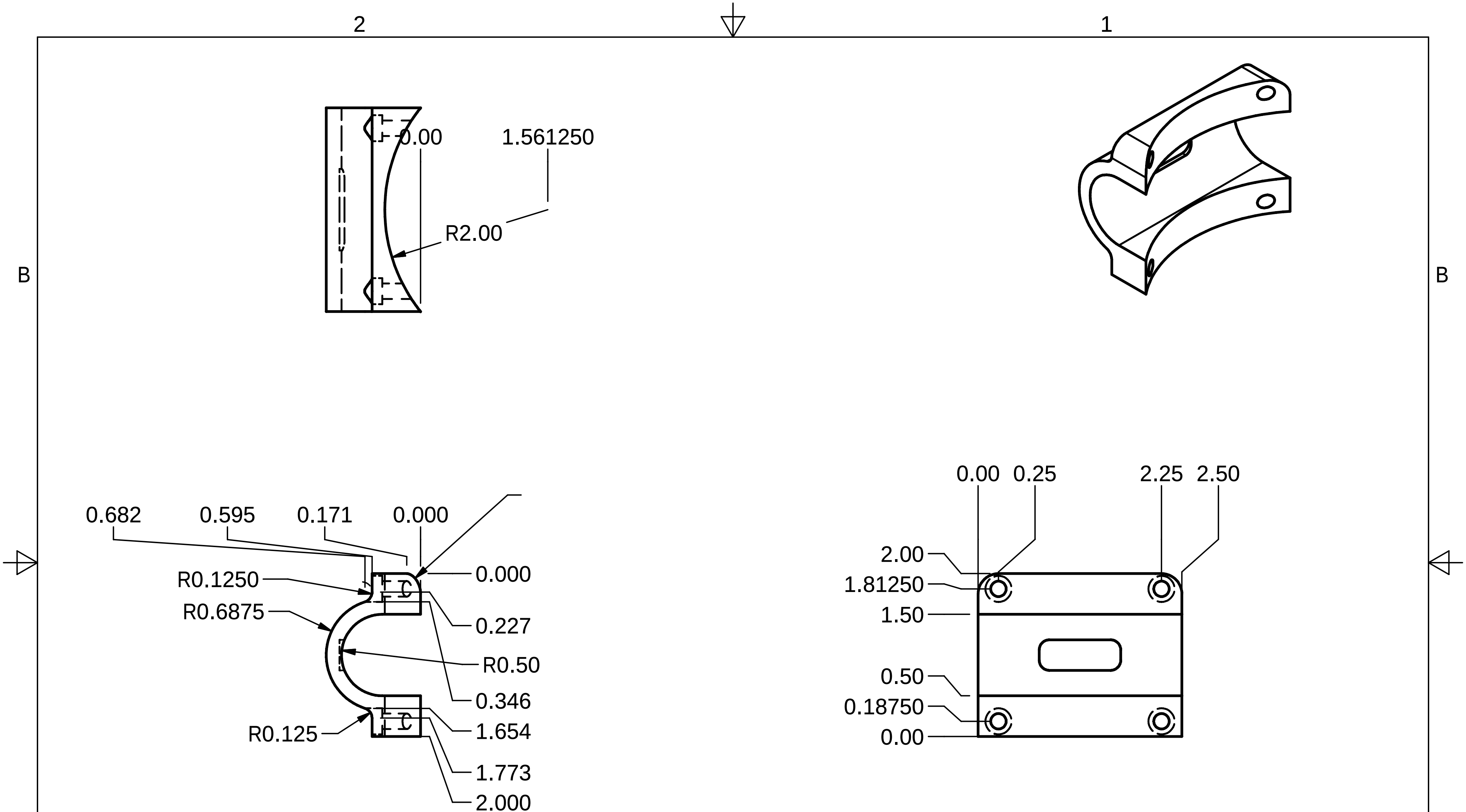


Figure 8 Pinion Housing Cap Layout Drawing



University of California, Davis Formula SAE			
TITLE Pinion Housing Cap			
DRAWN BY Jon		DATE 5/7/2006	REV
MATERIAL Aluminum-6061	UNLESS OTHERWISE SPECIFIED .XX ± .03 TOLERANCES ON MACHINE DIMS: .XXX ± .005		
QTY			
SIZE A	SCALE	DWG NO. / FILENAME pinion housing cap_v2.ipt	SHEET 3 of 3

Figure 9 Pinion Housing Cap Layout Drawing



University of California, Davis

Formula SAE

TITLE

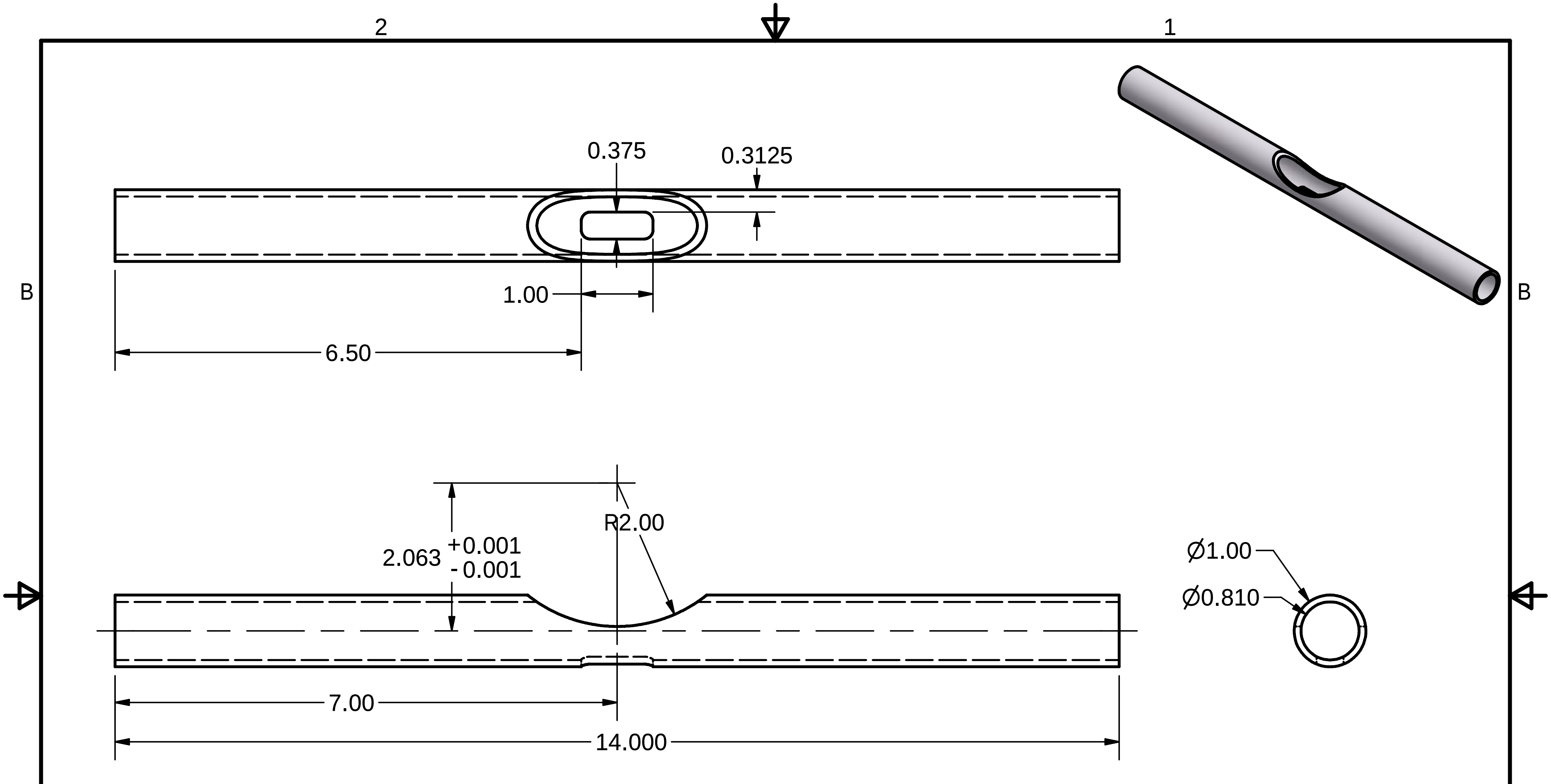
Pinion Housing Clamp

DRAWN BY	DATE	REV
Thomas	5/7/2006	

MATERIAL	UNLESS OTHERWISE SPECIFIED	.XX ± .03
Aluminum-6061	TOLERANCES ON MACHINE DIMS:	.XXX ± .005
QTY		

SIZE	SCALE	DWG NO. / FILENAME	SHEET 1 of 1
A		pionion housing clamp_v2.ipt	

Figure10 Pinion Housing Clamp Layout Drawing



University of California, Davis

Formula SAE

TITLE

Main Support Tube

DRAWN BY	DATE	REV
Jon	5/11/2006	

MATERIAL Aluminum-6061

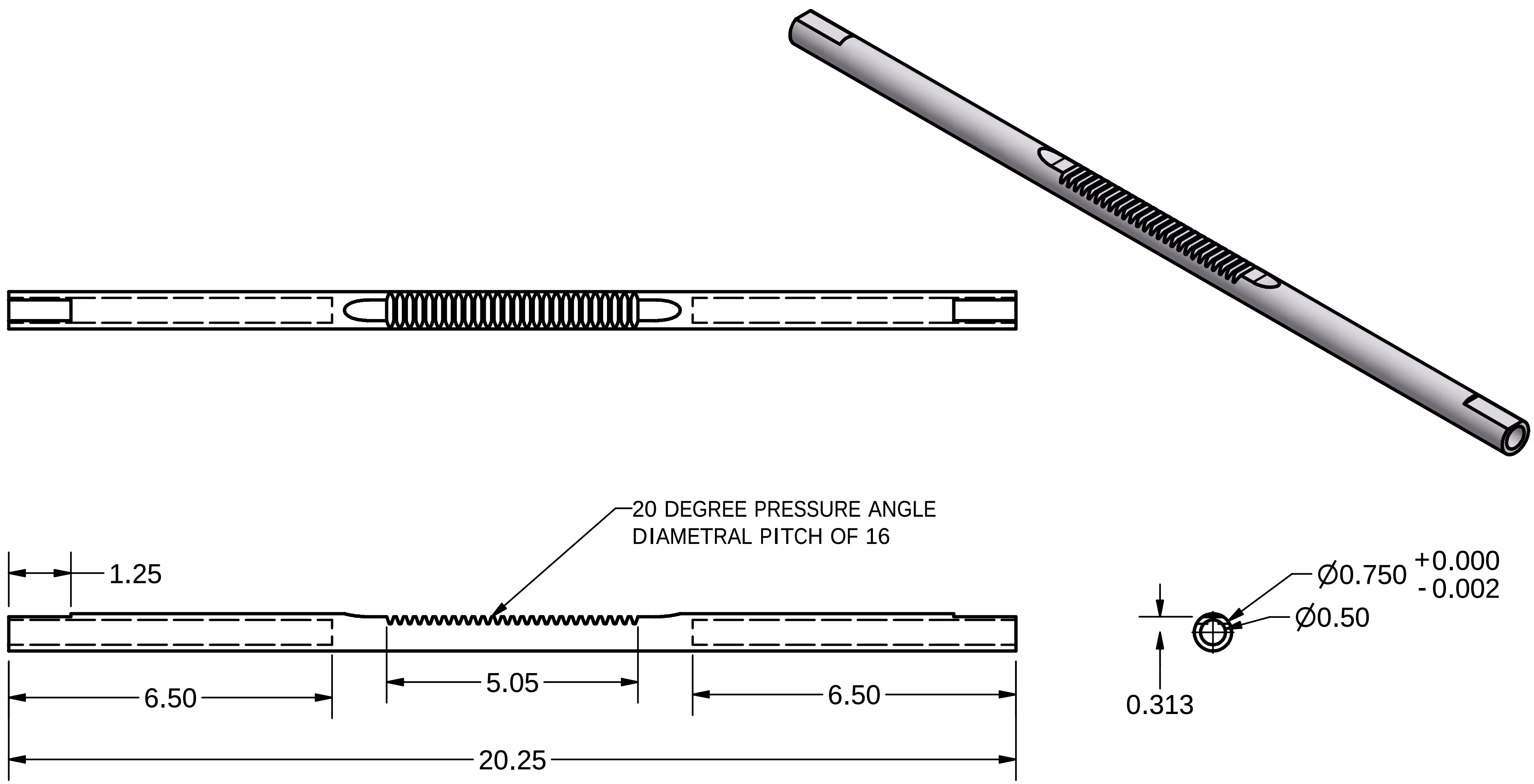
QTY

UNLESS OTHERWISE SPECIFIED .XX ± .03

TOLERANCES ON MACHINE DIMS: .XXX ± .005

SIZE	SCALE	DWG NO. / FILENAME	SHEET 1 of 1
A		tube_v2.ipt	

Figure 11 Main Housing Tube Layout Drawing



University of California, Davis Formula SAE			
TITLE Rack Gear			
DRAWN BY Jon		DATE 6/2/2006	REV
MATERIAL Aluminum-6061 QTY	UNLESS OTHERWISE SPECIFIED .XX ± .03 TOLERANCES ON MACHINE DIMS: .XXX ± .005		
SIZE A	SCALE	DWG NO. / FILENAME Final Rack2.ipt	SHEET 1 of 1

Figure 12 Rack Gear Layout Drawing

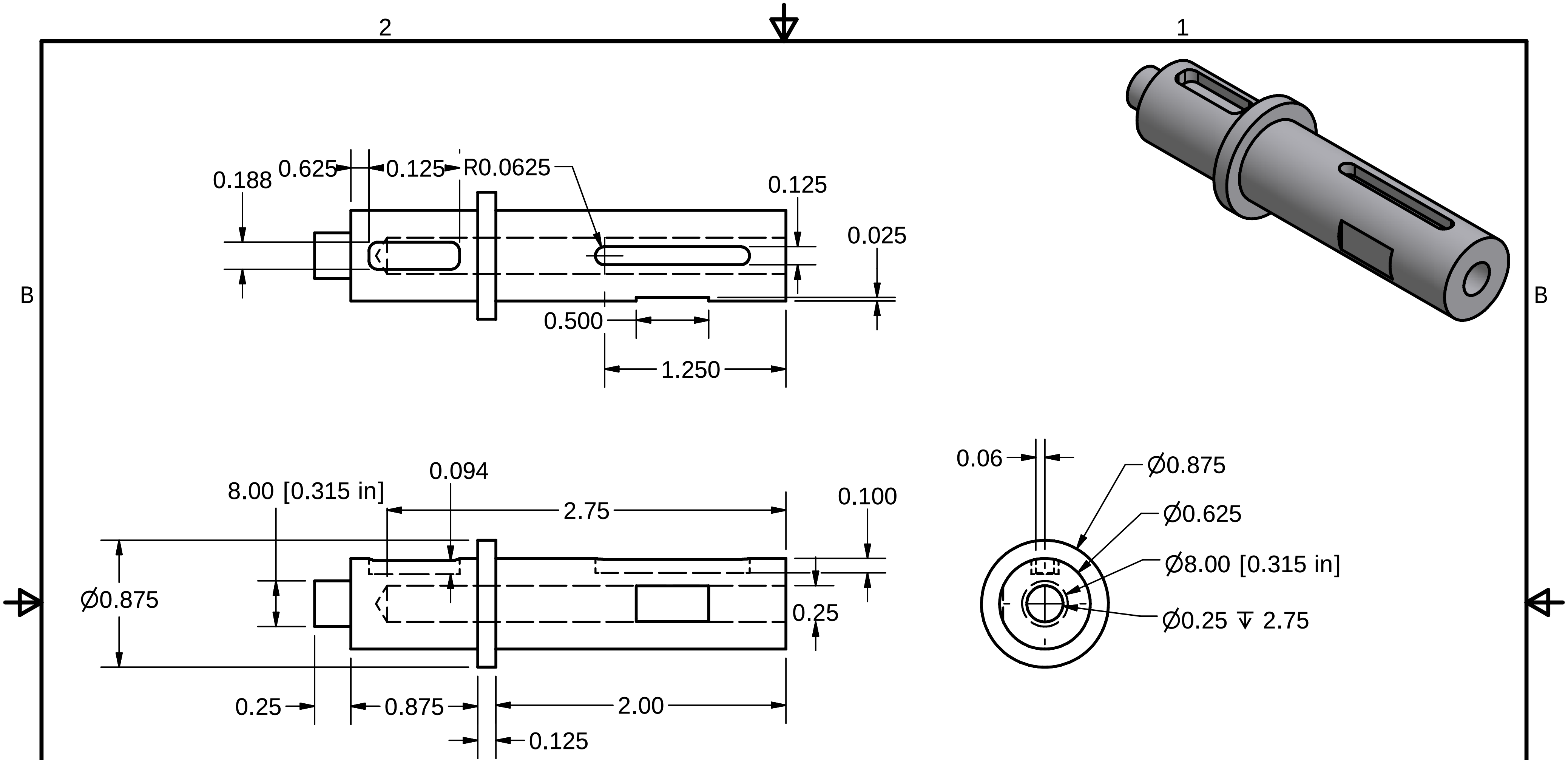


Figure 13 Upper Pinion Shaft Layout Drawing

University of California, Davis Formula SAE			
TITLE Upper Pinion Shaft			
DRAWN BY Jon		DATE 5/11/2006	REV
MATERIAL Steel, Mild	UNLESS OTHERWISE SPECIFIED .XX ± .03 TOLERANCES ON MACHINE DIMS: .XXX ± .005		
QTY			
SIZE A	SCALE	DWG NO. / FILENAME pinion shaft_upper_v2.ipt	SHEET 1 of 1

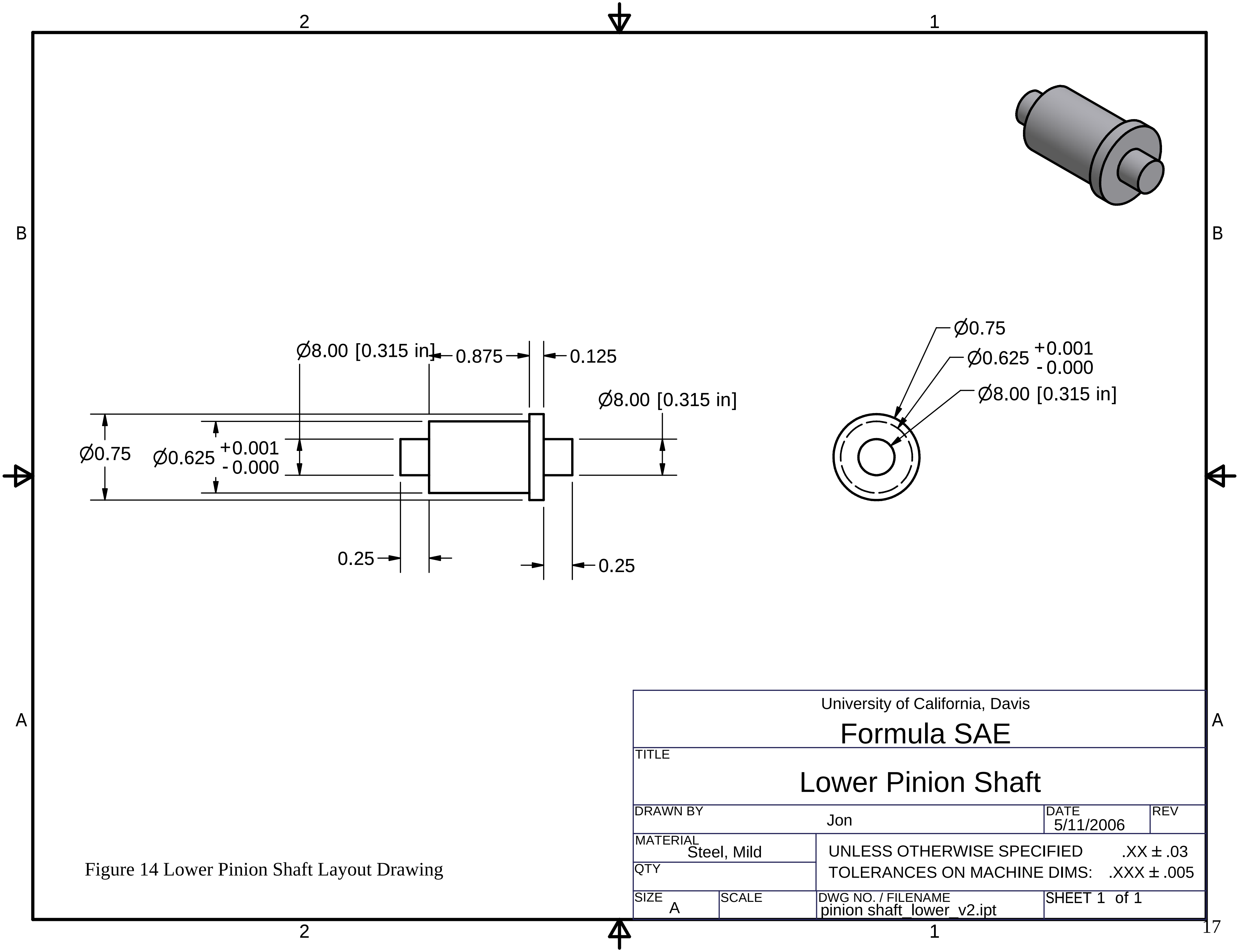
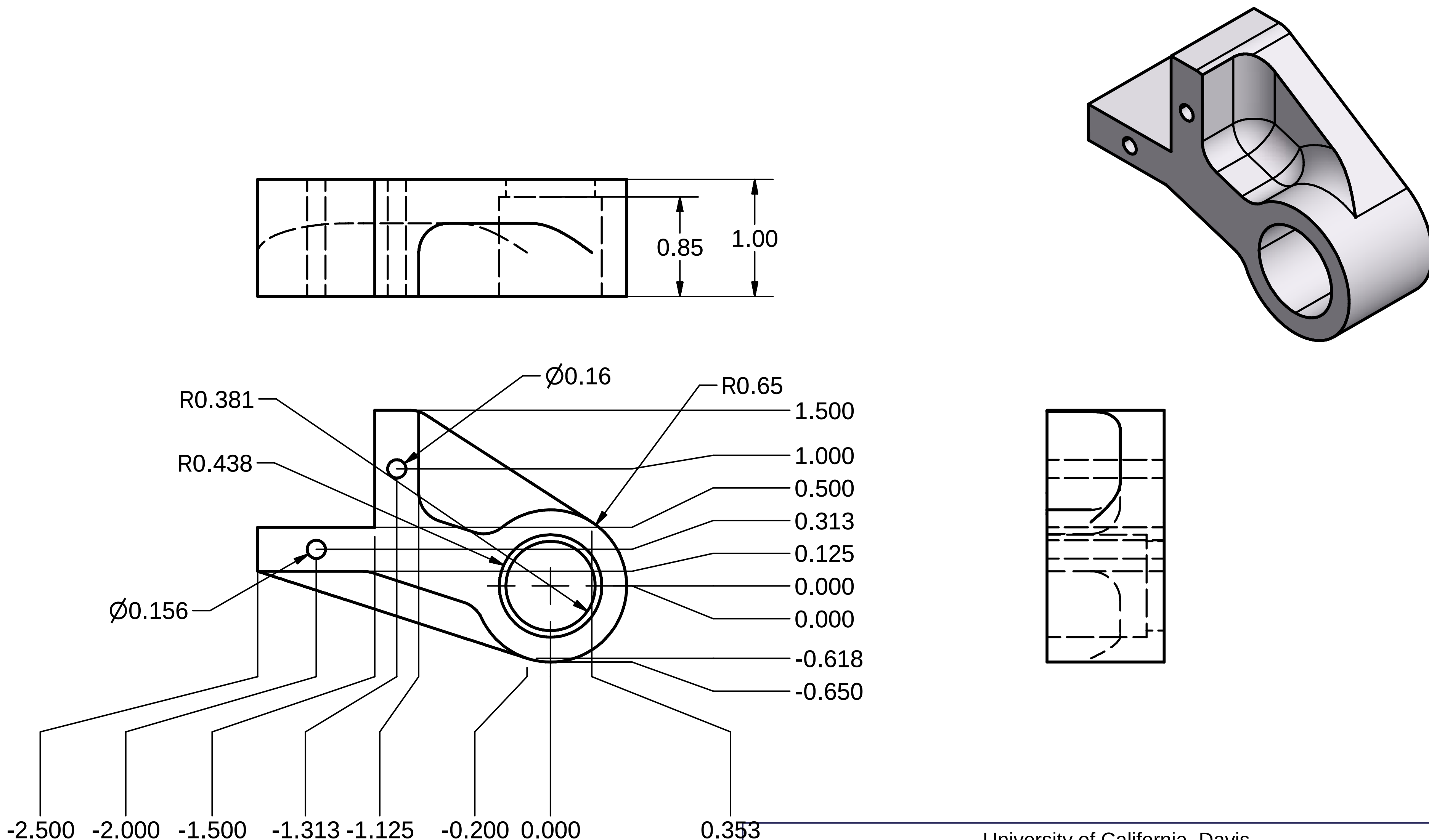


Figure 14 Lower Pinion Shaft Layout Drawing



University of California, Davis

Formula SAE

TITLE
End Support

DRAWN BY	DATE	REV
Jon	6/3/2006	

MATERIAL Aluminum-6061	UNLESS OTHERWISE SPECIFIED .XX ± .03
QTY	TOLERANCES ON MACHINE DIMS: .XXX ± .005

SIZE	SCALE	DWG NO. / FILENAME	SHEET
A		end support_final.ipt	SHEET 1 of 2

Figure 15 Housing End Support Layout Drawing

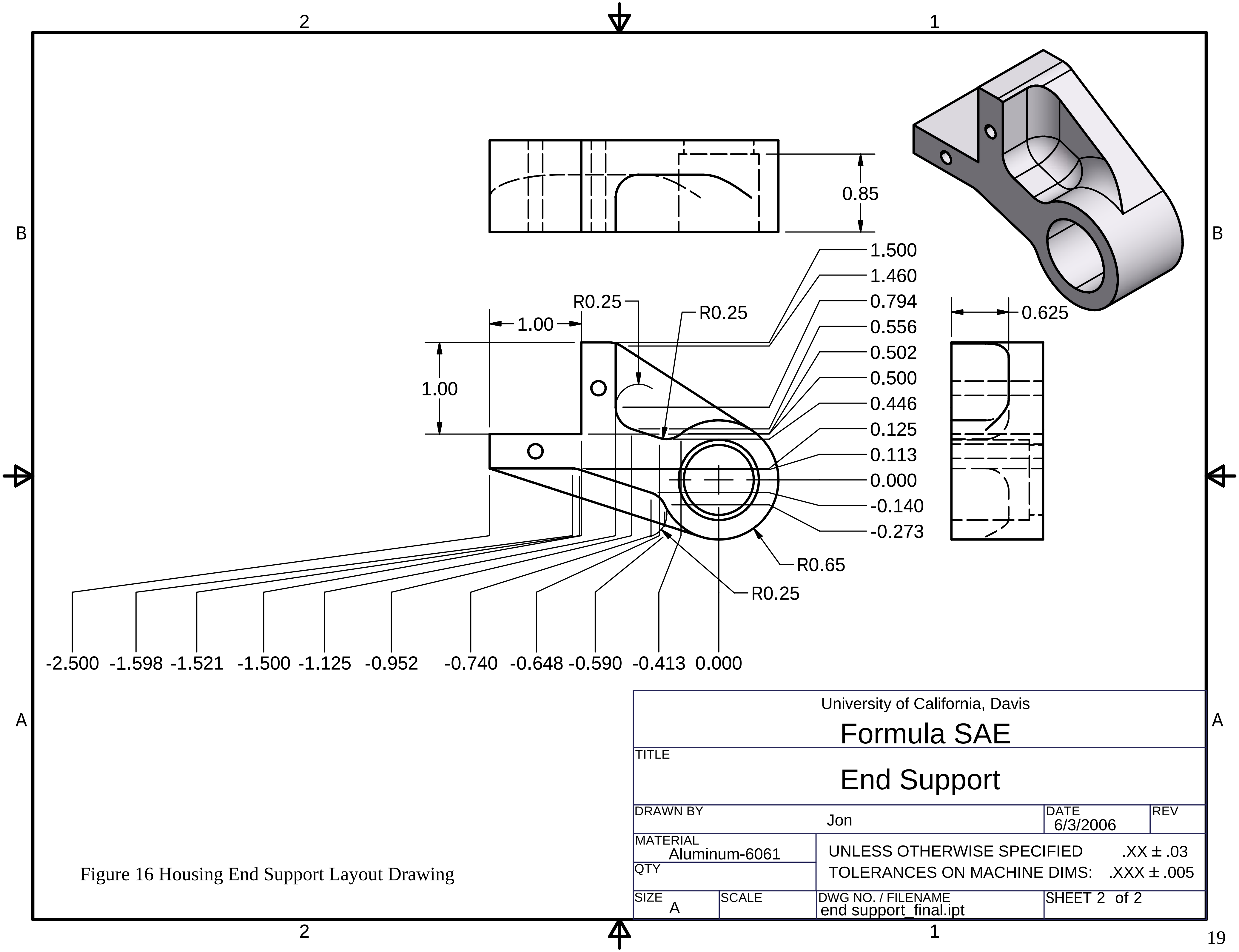
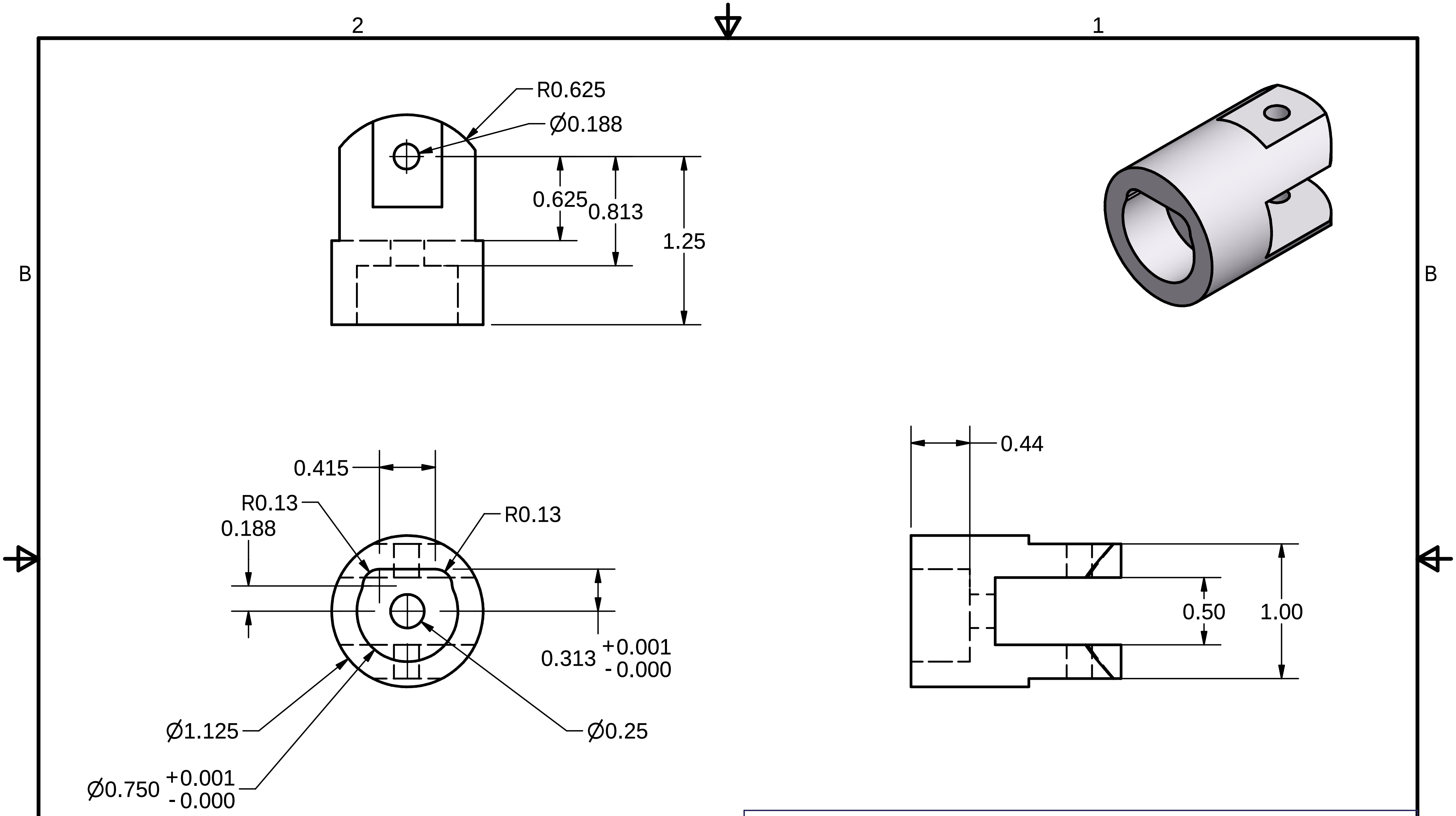


Figure 16 Housing End Support Layout Drawing



University of California, Davis

Formula SAE

TITLE

Rack End

DRAWN BY	Jon	DATE 6/2/2006	REV
MATERIAL Aluminum-6061	UNLESS OTHERWISE SPECIFIED .XX ± .03		
QTY	TOLERANCES ON MACHINE DIMS: .XXX ± .005		
SIZE A	SCALE	DWG NO. / FILENAME rackendfinal.ipt	SHEET 1 of 1

Figure 17 Rack End Layout Drawing

FINAL BILL OF MATERIALS

Although all manufacturing was performed in the student machine shop by our group members, the bill of materials includes estimated manufacturing costs. The entire prototype unit was created using a Bridgeport vertical mill with 3-axis Millpower CNC control, a Clausing Colchester engine lathe, band saws, grinders, and common hand tools. Production runs of this product would utilize full CNC mills and lathes, which, once set up properly, are capable of rapidly machining parts. Our machining time and labor time estimates assume manufacturing times representative of a production run.

A flat rate of $70/hr was assumed for CNC operations due to the expense of the machine, plant overhead, operator salary, and power consumption. Similarly, a flat rate of $35/hr was assumed for all other general labor, such as jigging, welding, and assembly work. This rate accounts for technician salary and plant overhead.

The actual cost for our team to construct the prototype unit is simply the total material cost shown in the BOM to be $141.36.

Sub System Name	Component Description	Purchased / Fabricated	Materials Needed	Parts Needed	Supplier	Part Number	Quantity	Unit	Cost per unit	CNC Machine Time [min]	Labor/Weld Time [min]	Material Cost	Labor and CNC Cost	Total Cost
Steering Rack	Rack Housing Tube	Fabricated	1" OD x 0.095" wall 6061-T6 Al tube		onlinemetals.com		14	in.	$0.35	0.5	0.5	$4.90	$0.88	$5.78
	Pinion Housing	Fabricated	2.5"x2.5" 6061-T6 Al square solid		onlinemetals.com		3	in.	$2.49	6.0	1.5	$7.47	$7.88	$15.35
	Pinion Housing Clamp	Fabricated	2.5"x2.5" 6061-T6 Al square solid		onlinemetals.com		1.5	in.	$2.49	4.0	1.5	$3.74	$5.54	$9.28
	Pinion Housing Cap	Fabricated	0.75"x 2.5" 6061-T6 Al bar		onlinemetals.com		3	in.	$0.86	2.0	0.5	$2.58	$2.63	$5.21
	Pinion Gears	Purchased		Boston Gear YB20 5/8	Motion Industries	ND15B	2	ea.	$11.95	0.2	0.1	$23.90	$0.29	$24.19
	Rack Gear	Fabricated	0.75" 1042 TGP steel solid bar		onlinemetals.com		20	in.	$0.46	16.0	0.2	$9.20	$18.78	$27.98
	Rack Ends	Fabricated	1.25" 6061-T6 Al solid rod		onlinemetals.com		5	in.	$0.57	3.0	0.5	$2.85	$3.79	$6.64
	Housing Supports	Fabricated	1"x 2" 6061-T6 Al bar		onlinemetals.com		6	in.	$0.94	3.0	0.2	$5.64	$3.62	$9.26
	Rack Gear Bushings	Fabricated	0.875" OD x 0.065" wall bronze tube		onlinemetals.com		2	in.	$0.80	0.8	0.1	$1.60	$0.99	$2.59
	Upper Pinion Gear Shaft	Fabricated	0.875" OD 4130 steel bar		onlinemetals.com		3.25	in.	$0.59	1.5	0.1	$1.92	$1.81	$3.73
	Upper Pinion Gear Keyway Stock	Purchased		3/16" Square Key Stock	McMaster-Carr	98830A150	0.75	in.	$0.06	0.0	1.5	$0.05	$0.88	$0.92
	Lower Pinion Gear Shaft	Fabricated	0.875" OD 4130 steel bar		onlinemetals.com		1.5	in.	$0.59	0.8	0.1	$0.89	$0.99	$1.88
	Pinion Gear Bushing	Fabricated	0.75" OD x 0.12" wall bronze tube		onlinemetals.com		0.625	in.	$1.02	0.2	0.1	$0.64	$0.29	$0.93
	Rack Gear Support Bushing	Fabricated	.5" x 1" bronze bar		onlinemetals.com		0.5	in.	$0.62	0.5	0.7	$0.31	$0.99	$1.30
	Pinion Gear Bearings	Purchased		22mm OD x 7mm ID skateboard bearing	Ground Zero	ABEC-3	3	ea.	$1.12	0.0	0.0	$3.36	$0.00	$3.36
	Pinion Housing Cap Bolts	Purchased		#8-32 x .5" Socket Head Cap Screw	McMaster-Carr	91251A194	7	ea.	$0.11	0.0	0.0	$0.77	$0.00	$0.77
	Pinion Housing Clamp Bolts	Purchased		#10-24 x .75" Socket Head Cap Screw	McMaster-Carr	91251A245	4	ea.	$0.10	0.0	0.0	$0.40	$0.00	$0.40
	Rack End Bolts	Purchased		1/4-20 x 1" Socket Head Cap Screw	McMaster-Carr	91251A542	2	ea.	$0.13	0.0	0.0	$0.26	$0.00	$0.26
	Housing Support Bolts	Purchased		1/4-20 x 2" Socket Head Cap Screw	McMaster-Carr	91251A550	4	ea.	$0.19	0.0	0.0	$0.76	$0.00	$0.76
	Sub System Assembly									0.0	12.0	$0.00	$7.00	$7.00
									Sub-total	$38.50	$19.60	$71.22	$56.35	$127.57

Table 1. Page 1 of Bill of Materials

Sub System Name	Component Description	Purchased / Fabricated	Materials Needed	Parts Needed	Supplier	Part Number	Quantity	Unit	Cost per unit	CNC Machine Time [min]	Labor/Weld Time [min]	Material Cost	Labor and CNC Cost	Total Cost
Steering Shaft	Shaft Tube	Fabricated	5/8" OD x .065" wall 4130 steel tube		onlinemetals.com		19	in.	$0.34	0.0	0.5	$6.46	$0.29	$6.75
	Quick Release Steering Hub	Purchased		Hub with 5/8" Hex Shaft	Chassis Shop	270-2017	1	ea.	$28.29	0.0	0.0	$28.29	$0.00	$28.29
	Steering Shaft Keyway Collar	Fabricated	1" OD x .1875" wall 4130 steel tubing		onlinemetals.com		3	in.		2.0	1.5	$0.00	$3.21	$3.21
	Steering Shaft Keyway Stock	Purchased		1/8" Square Key Stock	McMaster-Carr	98830A100	1.25	in.	$0.06	0.0	1.5	$0.07	$0.88	$0.94
	Steering Shaft Set Screws	Purchased		#10-32 x 5/6" socket set screw	McMaster-Carr	91375A439	2	ea.	$0.08	0.0	0.0	$0.16	$0.00	$0.16
	Sub System Assembly									0.0	5.0	$0.00	$2.92	$2.92
									Sub-total	$2.00	$8.50	$34.98	$7.29	$42.27
Tie Rods	Tie Rod Tubes	Fabricated	0.5" OD x 0.035" wall 4130 steel tube		onlinemetals.com		18	in.	$0.33	0.0	1.0	$5.86	$0.58	$6.45
	Tie Rod End Bungs	Fabricated	0.5" OD 4130 steel round bar		onlinemetals.com		4	in.	$0.17	1.2	0.3	$0.69	$1.58	$2.27
	Right Hand Thread Rod End	Purchased		0.19" bore x 1/4-28 shank spherical joint	Aurora	XAM-3	2	ea.	$6.27	0.0	0.0	$12.54	$0.00	$12.54
	Left Hand Thread Rod End	Purchased		0.19" bore x 1/4-28 shank spherical joint	Aurora	XAB-3	2	ea.	$6.89	0.0	0.0	$13.78	$0.00	$13.78
	Rod End Bolt	Purchased		#10-28 x 1.5" AN-3 Bolt	Aircraft Spruce	AN3-14A	4	ea.	$0.16	0.0	0.0	$0.64	$0.00	$0.64
	Rod End Washer	Purchased		#10 AN thin flat washer	Aircraft Spruce	AN960-10L	8	ea.	$0.03	0.0	0.0	$0.24	$0.00	$0.24
	Rod End Nut	Purchased		#10-28 half height nylon lock washer	Aircraft Spruce	AN364-428A	4	ea.	$0.13	0.0	0.0	$0.52	$0.00	$0.52
	Rod End Right Hand Thread Jam Nut	Purchased		1/4-28 RH thread thin check nut	Aircraft Spruce	AN316-4R	2	ea.	$0.12	0.0	0.0	$0.24	$0.00	$0.24
	Rod End Left Hand Thread Jam Nut	Purchased		1/4-28 LH thread thin check nut	Aircraft Spruce	AN316-4L	2	ea.	$0.32	0.0	0.0	$0.64	$0.00	$0.64
	Sub System Assembly									0.0	8.0	$0.00	$4.67	$4.67
									Sub-total	$1.20	$9.30	$35.16	$6.83	$41.98
									TOTAL	**$41.70**	**$37.40**	**$141.36**	**$70.47**	**$211.83**

Table 2. Page 2 of Bill of Materials

SUMMARY REPORT

BASIC DESIGN FEATURES

The steering system designed for the 2006 UC Davis Formula SAE car is a rack and pinion type design. Although more expensive and complex than some designs, a rack and pinion steering system provides the greatest precision and packaging. Since the team is on a strict budget, cost is a factor but only if it does not compromise vehicle performance. The overall vehicle goals target quick acceleration and responsive cornering abilities. Low weight and proper suspension geometry became the governing factors in steering system design, provided the system stayed within the team budget. Manufacturability, ergonomics, low backlash, and chassis constraints were all high priority secondary considerations. Shown in Figure 18 is the final assembly model of our steering system.

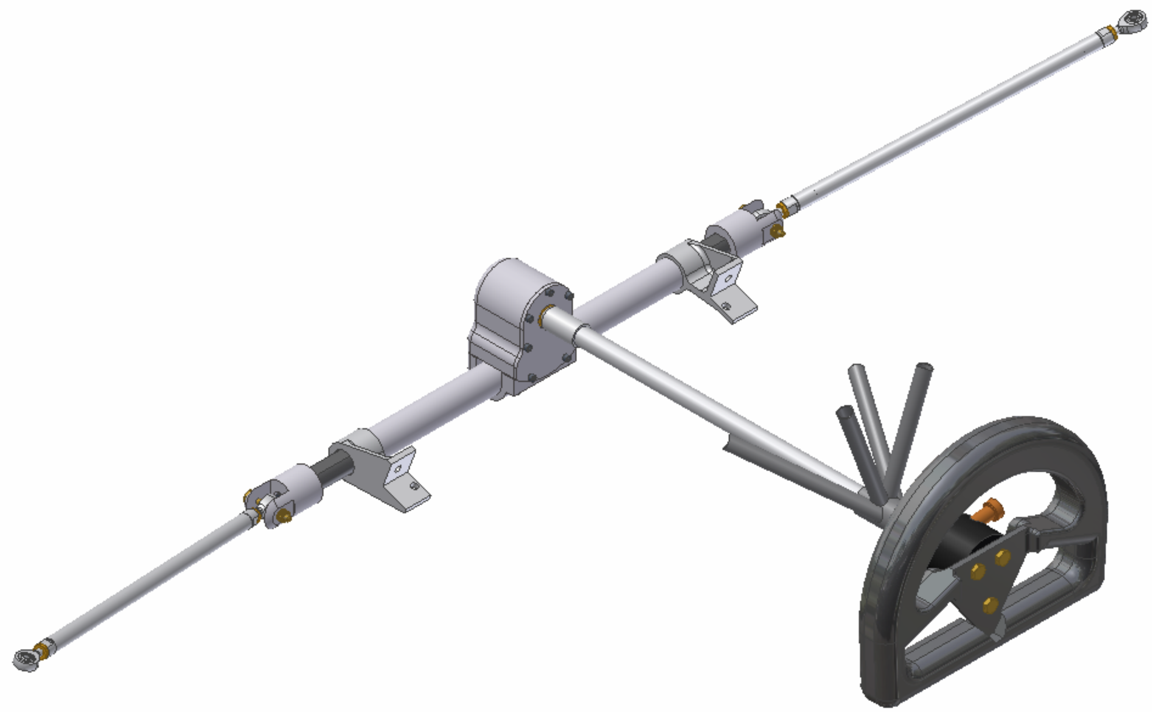


Figure 18. Complete Steering System Model

The steering system is comprised of three major sub-systems; the steering shaft, the steering rack, and the tie rods. The steering shaft connects the steering wheel to the steering rack input shaft and is manufactured from 4130 steel tubing. A keyed collar with set screws firmly attaches the two components together while still allowing removal for service.
The steering rack, shown in Figure 19, is by far the most complex and critical element of the system. A multiple piece housing contains rack and pinion gears and locates them relative to each other through a series of roller ball bearings and bronze bushings. The steering rack mounts to the vehicle's chassis through two machined aluminum mounts located at the ends of the steering rack housing for greatest possible support.

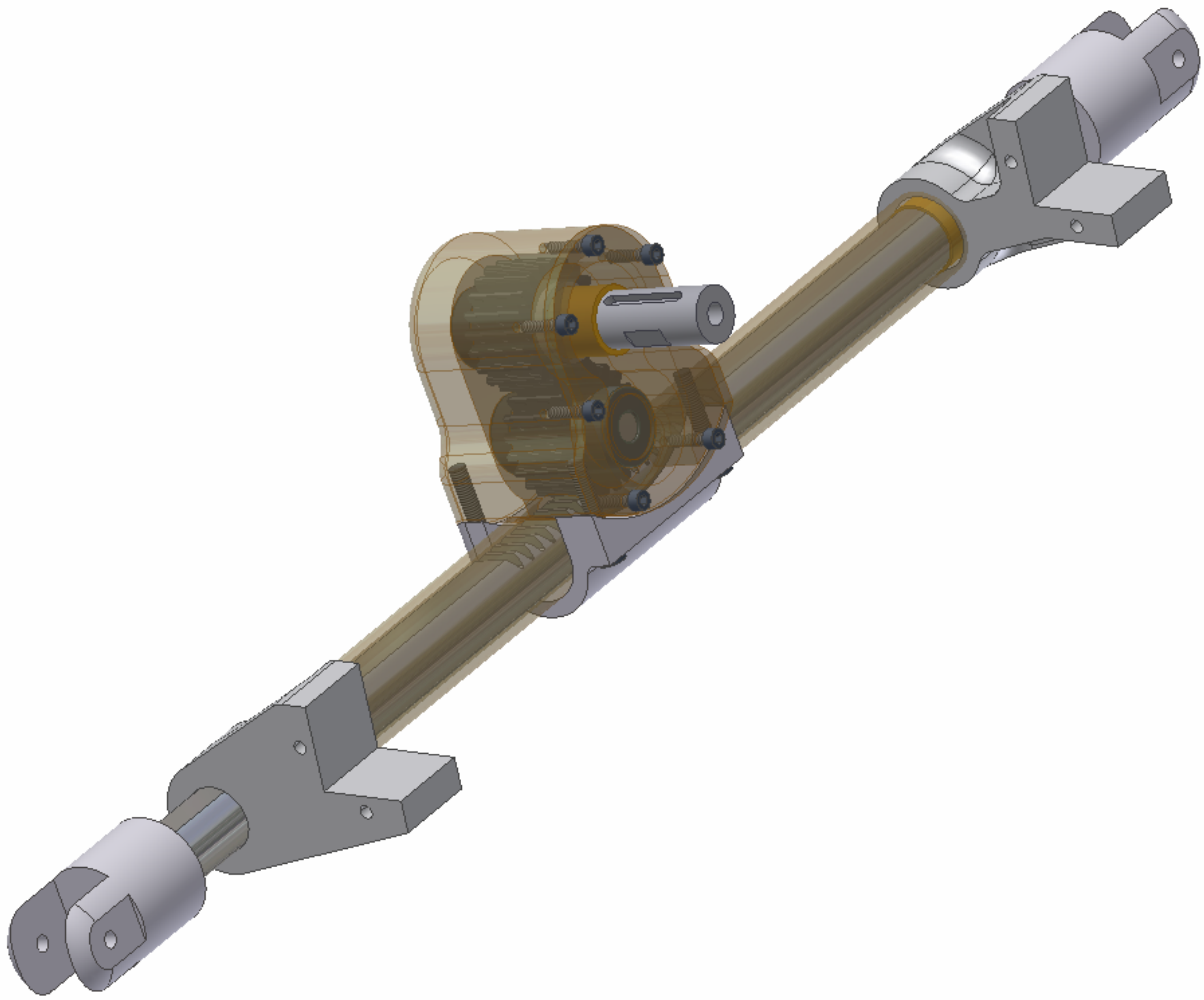


Figure 19. Steering Rack Model

The tie rods are responsible for connecting the steering rack to the front uprights. This connection transmits input forces from the steering rack to the steering arms and causes a rotation of the wheels around their steering axis. The tie rods are constructed of 4130 steel tubing and have threaded bungs welded into the ends for spherical rod ends the thread into. The rod ends are secured with jam nuts to prevent any unwanted length adjustment, resulting in improper wheel alignment.

Although the steering system designed is a fairly standard rack and pinion system, there are multiple unique features that make the system ideal for the intended application. The dual pinion design, which the team has never seen implemented, allows the pinion housing to be placed above the rack centerline. The additional leg clearance from moving the housing above the rack provides increased driver comfort and easier ingress and egress from the vehicle. This results in a safer, more comfortable vehicle. The dual pinion design also has another benefit. Due to the additional height and placement of the pinion housing, a straight steering shaft with no flex joints is possible. In past cars, the flex joints in the steering shaft have always been a source of slop in the steering system. By eliminating these joints without compromising steering wheel orientation, the overall precision of the system has been drastically increased.

One of our critical design elements was 1 piece rack gear that connected the two tie rods together. Long square rack gears are commonly available, but the square gear would no have worked well in this design. A round rack gear with gear teeth only in the middle portion provided a clean and simple way to support the gear. Two bronze bushing securely locate the gear and support non-axial loads transmitted to the gear. Since no commercially available rack gears met our specifications, a custom rack gear was made in house using a CNC wire EDM process. The resulting gear was ideal for our application and was a perfect match to the pinion gears selected. Additionally, the surface finish from the EDM process is superior to the gear cutting process used by most manufacturers.

The final key element of the design is the mounting style connecting the rack housing to the chassis. Unlike the commercially available design that has a single mount at the center of the rack housing, our design utilizes two outboard mounts placed at each end of the rack housing. By supporting the steering rack in this manner, deflection due to steering loads is nearly eliminated, reducing steer compliance and increasing driver control.

DESIGN FUNCTIONALITY

The steering system converts rotational input provided by the driver through the steering wheel to linear motion. This linear motion exerts a force on the steering arm through the tie rod, which creates a moment about the steering axis. The steering arm is fixed to the suspension upright, so the result is a change in steer angle of the tires and wheels. By turning the tires relative to the direction of travel, a slip angle is created in the tire, where it contacts the ground, known as the contact patch. This slip angle is actually a small deformation of the rubber in the tire at the contact patch due to the imposed steering angle. The result is a generation of lateral force between the tire and the ground, allowing the vehicle to change yaw angle and maneuver through a course.

Descriptions of the components that comprise the steering rack assembly are given bellow along with details about how the parts interact to form a functioning rack and pinion system.

Rack Housing:

The steering rack housing is the most critical subsystem in the steering system for the Formula SAE car. The rack housing locates the rack and pinion gears, supports the loads generated from the gears, and distributes loads from the tie rods to the chassis. The design for the rack housing utilizes four major components; a main support tube (blue), a pinion gear housing (purple), a pinion gear housing clamp (green), and a pinion housing cap (red). The assembled steering rack housing can be seen in Figure 20, showing the orientation of the four components.

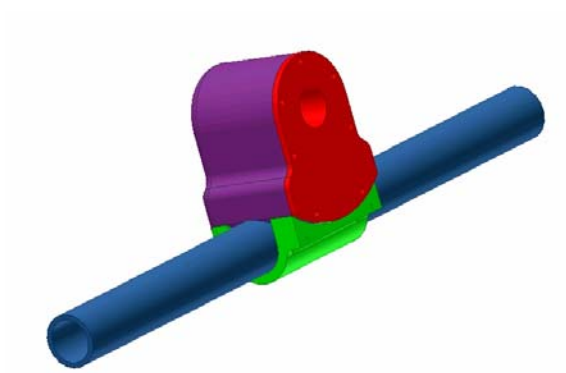


Figure 20. Steering Rack Housing

Main Support Tube:

The main support tube for the rack housing, shown in blue in Figure 20 and with the rack gear in Figure 21, is the main member of the steering rack housing. The tube is manufactured from 1” OD x 0.95” wall 6061-T6 aluminum tube. As can be seen in Figure 21, a notch is milled into the tube in order to expose the teeth on the rack gear and allow the rack and pinion gears to mesh. In addition to exposing the rack gear, this notch was designed to precisely locate the pinion housing (purple in Figure 20). There is a bronze bushing located in each end of the tube with a press fit to support the rack gear. Any non-axial loads exerted on the rack gear are transmitted through these bushings to the tube.

Figure 21. Main Support Tube

Pinion Housing:

The pinion housing, as the name suggests, contains the pinion gears as well as the bearings for the pinion gears. This component is shown in purple in Figure 20 and with the pinion gears in Figure 22. The curve on the bottom side of the housing, seen in Figure 22, mates with the notch cut in the main support tube. When these two surfaces are held in contact, the lower pinion gear and the rack gear are maintained in proper mesh. The slot machined on the bottom side provides clearance necessary for the rack gear. Two bearing bores are machined inside the housing to accommodate 22mm ball bearings that will allow the pinion gears to rotate freely under load. These bearing bores are shown clearly in Figure 23.

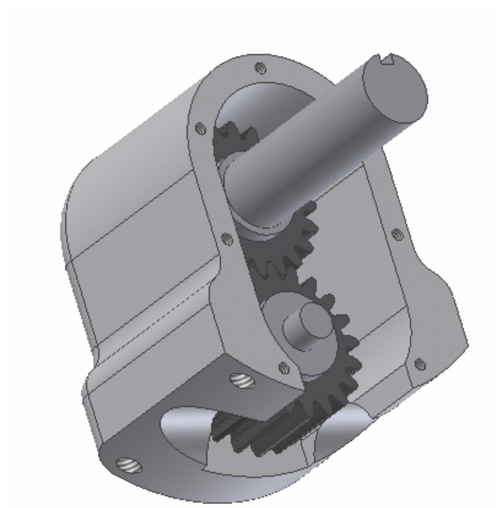


Figure 22. Pinion Housing with Gears

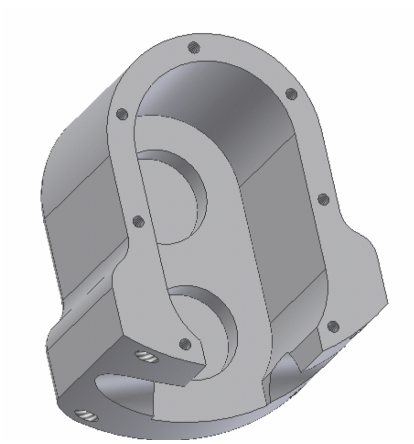


Figure 23. Pinion Housing

Pinion Housing Cap:

In order to allow the gears and bearings to be installed in the pinion housing, a removable front plate was needed. This plate, shown in red in Figure 20 and the backside of it in Figure 24, is fastened to the pinion housing with seven #8-32 bolts. The lower pinion gear uses another 22mm ball bearing, housed in the support cap, to support the other side of the gear shaft and prevent deflection. The upper pinion gear, however, must include a larger shaft for the steering shaft to connect and transfer torque input from the driver. In order to allow this, the upper pinion uses a large, 5/8" ID bronze bushing on the cap side for support. This design allows a 5/8" shaft to protrude through the cap and meet the steering shaft. A larger bearing was considered for this location, but the additional weight from the oversized bearing was too large to justify when compared to the bushing.

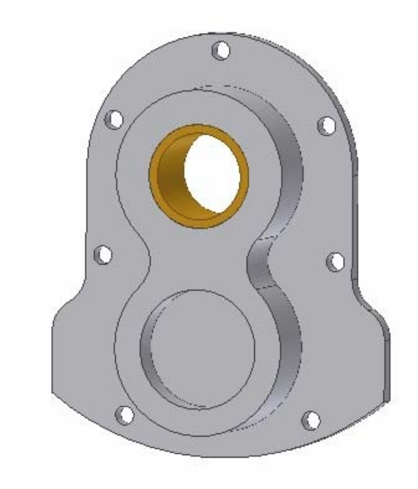


Figure 24. Pinion Housing Cap

Pinion Gear Housing Clamp:

The key to ensuring that proper gear mesh is maintained between the pinion and the rack gear is that the pinion housing is held tightly to the main tube. This is accomplished through the use of the pinion housing clamp, shown in green in Figure 20 and separately in Figure 25. The clamp wraps around the bottom of the main tube and utilizes four #10 bolts to secure the pinion gear housing in place. The curved mating surface between the pinion housing and the tube, combined with the clamping force, eliminates any possible movement of the pinion gear housing in all three planes and rotation on all three axes.

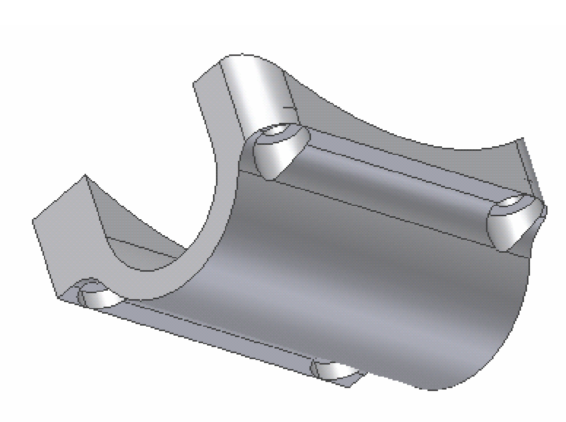


Figure 25. Pinion Gear Housing Clamp

When the driver exerts a torque on the steering wheel, the torque is transmitted through the steering shaft to the upper pinion gear. The torque is then transmitted though the lower pinion gear to the rack gear, where it becomes a lateral force. In order to cause a linear motion of the rack gear, the pinion housing must withstand the resulting lateral reaction force. The curved notch cut in the main tube and the corresponding curve on the bottom of the pinion housing (Figures 21 and 23), along with the clamp, allow the pinion housing to provide this force by preventing the housing from sliding along the axis of the tube. In order for the housing to slide along the tube, the entire pinion housing would be forced to move away from the tube centerline due to the curved profile of the notch. The pinion housing, however, is firmly held in relation to the centerline of the tube by the clamp, meaning that any relative lateral motion between the tube and the pinion housing is prevented.

Rotation around the tube axis is also prevented again by the pinion housing being held tightly to the flat surface created by the notch in the tube. As long as the housing remains in contact with the tube surface, any rotation around the tube axis is not possible.

Pinion Gears:

From experience with previous FSAE cars, it was determined that a steering ratio of 3:1 for steering wheel rotation to steer angle was ideal. This means that in order to steer turn the wheel to full lock of 30 degrees in one direction, the steering wheel would have to be turned 90 degrees. Due to the extremely tight courses that are encountered at competition, this quick steering ratio has proven to be ideal. The driver never has to remove or even re-position their hands on the wheel for the entire steering range. Using SusProg 3D, a suspension design program that was used to design the suspension system on the 2006 FSAE car, we were able to determine that the rack must move 0.975” laterally to cause a 30 degree change in steer angle. In order for a 90 degree rotation of the steering wheel to result in the desired 0.975” linear motion of the rack gear, a pinion gear with a pitch diameter of 1.241 would be necessary. The closest commonly available pitch diameter is 1.25” and should work quite well for our application. The steering ratio will be slightly lower than desired, but the difference should be imperceptible. Final gear choice was a Boston Gear YB20-5/8 pinion gear, which is a 20 degree pressure angle, 20 tooth, steel pinion gear with a diametral pitch of 16, and a ¾” face width. A model of this pinion gear can be seen in Figure 26. The gears come with a large hub and set screw. In order to reduce overall size of the pinion housing, these hubs were machined off the gears.

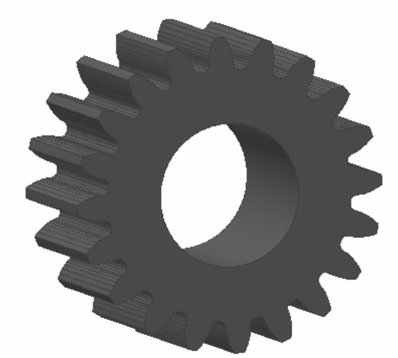


Figure 26. Pinion Gear

Pinion Shafts:

The pinion gears are mounted on custom steel pinion shafts, that are designed to locate the gear and interface with the bearings and bushing. The upper and lower pinion shafts can be seen in Figures 27 and 28, respectively. The upper shaft is turned down to 7mm at one end to fit the ID of the bearing and the other end remains at 5/8” to mate with the bronze bushing in the pinion housing cap. The shaft extends an additional 1.5” and has a keyway machined into to it to accept the steering shaft, connecting the steering wheel to the upper pinion. The gear side of the shaft also has a keyway machined into it to help in torque transfer from the shaft to the gear. The pinion gear was placed over the shaft and keyway with a slight press fit and then TIG welded onto the shaft. The lower shaft is similar in design to the upper shaft, but unlike the upper, provisions are needed for steering shaft connections and no keyway is machined for the gear. The reason for this is because there is no torque that must be transmitted from the

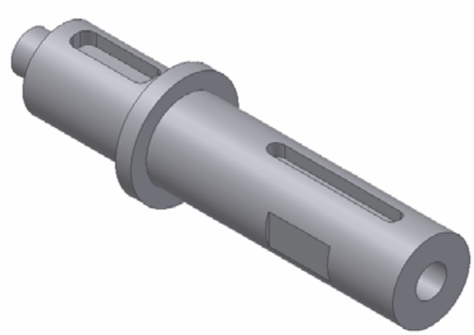


Figure 27. Upper Pinion Shaft

Figure 28. Lower Pinion Shaft

gear to the shaft as it is simply an idler gear.

With no steering shaft connection needed, a bearing could be used to support each end of the shaft. One bearing was pressed into the pinion housing and the other was pressed into the pinion housing cap. Figure 28 clearly shows how each end of the shaft is turned down to 7mm in order to fit the two bearings. The lower pinion gear was also be pressed onto the shaft and TIG welded in place. Figure 29 clearly illustrates the orientation of the upper and lower and shafts, along with the pinion gears in place.

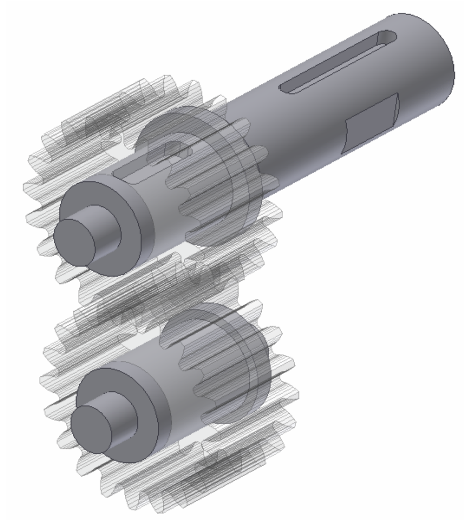


Figure 29. Pinion Shafts and Gears

Rack Gear:

Due to the very specific criteria of our rack gear, it was determined that a custom gear was the only option after the search for a suitable gear was unsuccessful. The mounting points for the tie rods must be 22 inches apart from each other so that the tie rods move in the same arc as the suspension arms, eliminating undesirable bump steer. The rack housing can only 14 inches wide due to chassis constraints, so that leaves 4 inches of the rack gear on each side unsupported. Additionally, the tie rods are not aligned directly with the axis of the rack gear. Under the worst case conditions, there is a 20 degree difference in angle between the tie rod and the rack gear. This necessary misalignment results in a component of the force being transmitted through the tie rod to act perpendicular to the rack gear axis. Excessive deflection in the rack gear would result in imprecise steering control and response. Calculations were performed to optimize rack gear material and diameter so that weight could be minimized without a large sacrifice in deflection. When considering cost, weight, strength, and material compatibility, it was determined that steel was the optimal material for the rack gear. Although initial thoughts were to use aluminum for the rack gear, concerns about galling between the steel pinion gear and aluminum rack gear resulted in the switch to steel. The steel chosen for the rack gear material was 1042 TGP, which is cold rolled, turned to shape, ground, and then polished. This material is very accurate dimensionally and has an excellent finish, which reduces friction between the rack gear and the bronze support bushings. A 0.75” diameter rack gear provided the optimal gear mesh and calculations determined that a 0.5” hole could be bored into each end of the rack gear to reduce weight while still providing sufficient strength. The weight of the rack gear was reduced from 2.52 lbs to 1.74

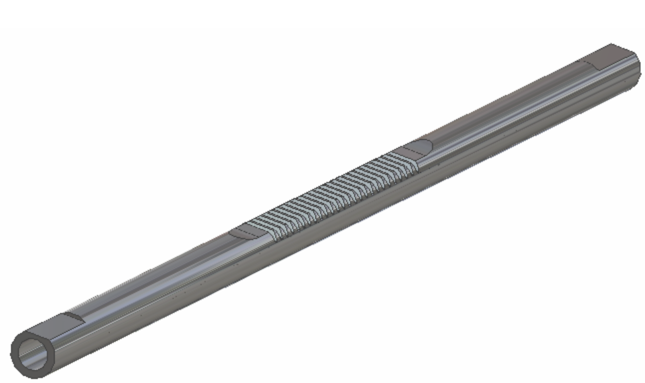


Figure 30. Rack Gear

lbs by drilling the two 6.5” long 0.5” holes in each end of the rack gear. The end result proved to be only 0.55 lbs heavier than the solid aluminum gear that was originally desired. Under maximum loading conditions, the deflection at the end of the drilled rack gear was found to be 0.00319” in bending and 0.000795” axially, a very acceptable amount. The results of these calculations, which were performed in Excel, can be seen in Appendix 3.

Once the diameter and material had been determined, the rack gear was designed. The teeth on the gear had to match the teeth on the pinion gears that were chosen. This meant the rack gear must also have a diametral pitch of 16 and a pressure angle of 20 degrees. The correct involute gear profile properties were calculated and then the teeth profile was modeled in Autodesk Inventor. The rack gear design can be seen in Figure 30. Based on the calculations from the pinion gear, the rack travel was known to be 0.975 inches in either direction. In order to ensure that the limiting factor on rack travel would not be the lack of teeth, the tooth section was designed to be 4 inches long and located at the center of the rack laterally. The final element to the rack design was provisions for the rack ends to bolt onto. The small flat section machined into each end of the rack allows the rack ends to index onto the rack and prevent any rotation. A single ¼-20 bolt is used to fasten the rack end to the rack gear, which had a small threaded bung welded into each end after the drilling process.

Rack Ends:

The rack ends, which attach to the ends of the rack gear, provide mounting points for the tie rod ends. The rack ends were machined from 6061-T6 aluminum bar, as it provides adequate strength and is inexpensive. The rack end is shown in Figures 31 and 15. A D-shaped slot was milled in the cylindrical rack end for a tight indexed fit on the rack gear (Figure 32). This indexing will prevent undesired steering geometry changes and binding which would result if the rack ends were to rotate. A through hole in each rack end allows a ¼-20 bolt to fasten the rack end to the threaded bung in the rack gear. The rod end will bolt into a machined C-channel to provide a double–shear mounting location.

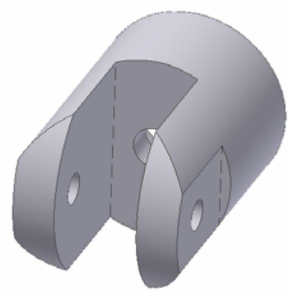


Figure 31. Rack End

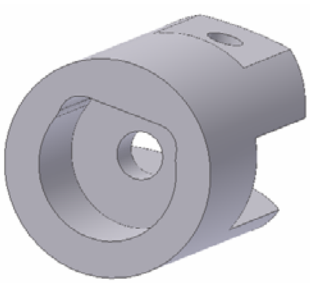


Figure 32. Rack End

The rack ends will also be used as steering stops. The maximum steering angle is controlled by the rack ends contacting the rack housing. A hard rubber stop will be placed between the housing and the rack ends to absorb impact forces and prevent damage. The thickness of these rubber stops can be changed to allow for specific maximum steering angles.

Housing End Supports:

The steering rack housing end supports are designed to accurately mount the assembly in the chassis with minimal deflections and minimal weight. Accurate mounting is necessary to set ideal steering geometry. Under loading the design needed to minimize lateral, forward, and angular deflections in order to maintain the correct steering geometries.

The rack housing supports achieve all of these design goals simultaneously. Forces are translated through the suspension and into the steering rack through the tie rod. The tie rod angle is relatively small, so the loading is primarily in the lateral direction. The aluminum T-beam design, shown in Figure 33, provides high strength per weight in the lateral direction and is easy to manufacture. To minimize angular deflection of the rack housing, the rack end mounts encase a wide segment of the rack housing tube. This minimizes angular deflection and localized stresses on the rack housing when forward loading is applied to the rack gear. Once the bronze bushings for the rack gear are pressed into the housing tube, the rack housing supports will be lightly pressed and bonded onto the tube. The bonds solidify the assembly and will not see significant loading. They merely prevent any rotation of the entire assembly around the rack axis.

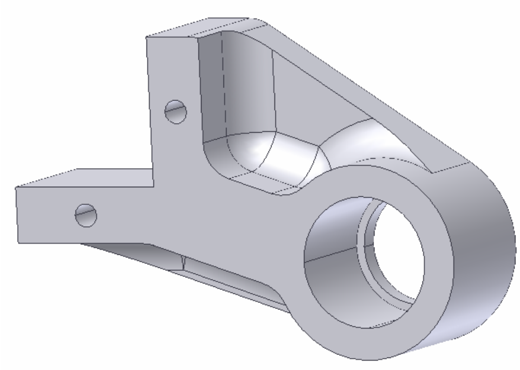


Figure 33. Housing Support

Two small tabs are welded onto the chassis located on each side of the supports. With this design, all lateral forces are transmitted directly to the chassis without stressing the mounting bolts. Additionally, this mounting style places the weld for the tabs in shear, which is far more ideal than tension or compression. The only tensile or compressive force seen by the welds is from non-axial loading on the rack gear, which is minimal compared to the lateral forces.

POSITIVE DESIGN ATTRIBUTES

The steering rack design incorporates attributes which have resulted in an accurate, manufacturable, and light weight assembly. The rack tube is machined from one piece of aluminum preventing binding and alignment problems. A multi-piece rack tube was used in a previous design which resulted in poor steering accuracy and feel. This rack tube is mounted to the vehicle chassis using two mounts on the rack tube ends. The location of these mounts reduces over all system deflection while maintaining the desired steering geometry. The rack housing, which contains the pinion gears, is bolted together preventing the need for welding. Not only does this provide easy assembly and maintenance but it prevents possible distortion from welding. An important aspect of all components is that they can be manufactured using basic machine tools.

DESIGN WEAKNESSES

Upon manufacture and assembly a few weakness in the design were discovered. One of these weaknesses is the junction between the pinion housing and pinion housing cap. The pinion housing cap is located on the pinion housing using only the mounting bolts. In this arrangement, the loads are supported solely by the bolts and not the housing material. To remedy this weakness a flange could be machined in the pinion housing to exactly locate the pinion cap. The other possible design change could be to use an aluminum rack gear. A steel rack gear is chosen to prevent galling and maintain a long service life but increased overall rack weight. More research into possible aluminum surface treatments would be required in order to achieve the desired performance from an aluminum rack gear.

REMAINING WORK AND DESIGN IMPROVEMENTS

Further work on the steering system will include various testing stages, many of which will be from driving the car. This will provide real user input as to how well the system works in its intended application. Once the system has been proven in the car, more improvements can be made to enhance overall optimization.

In future iterations of the gears, a more in depth analysis of metal hardening processes could be performed to allow for the use of aluminum a pinion or rack material. As a result of hardening one of the gears, we will be striving to reach a incompatibility between the metals of rack and pinion gears. This incompatibility is to ensure that contact between the gears will not cause an alloy forming reaction. A formation of alloys directly contributes to the galling of the gear teeth. One possibility might be hard anodizing which could potentially eliminate the incompatibility. While there are presumably many aerospace options that could be investigated, the overall cost of production and target market must still be considered.

Another improvement could be the use of a self locating pinion housing cap. In the current design, the cap is solely located by the bolts fastening it to the pinion housing. While this is effective and does not negatively affect the performance of the steering, the self-locating design has the capacity to potentially improve the pinion shaft alignment. The number of bolts could then also be re-evaluated to help with reduce weight and reduce assembly time. The case of the self locating pinion housing cap would also eliminate any shear stresses on the bolts as the alignment provision would put them purely in tension.

A final improvement which could be easily integrated into the steering system design is a steering force data acquisition system. This would utilize a rotary encoder and strain gauges. The rotary encoder would be implemented by re-manufacturing the upper pinion shaft and extending it through the forward face of the pinion housing. Strain gauges could then be fixed on the tie rods. Using the rotary encoder and the strain gauges would allow the collection of data relating steering forces to steer angle. From this data, optimization could be further performed on the components of the steering system.

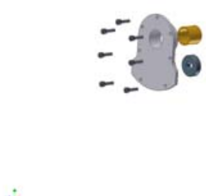
Assembly Process

Step 1
Press both rack bushings into each end of the tubular rack housing.

Step 2
Slide the rack gear into the housing, ensuring that it is centered.

Step 3
Press the shaft bearings into the pinion housing, and install both shafts.

Step 3
Press the idle shaft bearing and input shaft bushing into the pinion housing cap, and bolt to the pinion housing.



Step 4
Mount the pinion housing on the rack housing with the pinion housing clamp and cap screws.

Step 5
Slide both housing supports onto each end of the rack housing.

Step 6
Slide each rack end onto the rack gear, and bolt in place. Ensure that the clevis slot is vertical.

Step 7
Install the final assembly into the car.

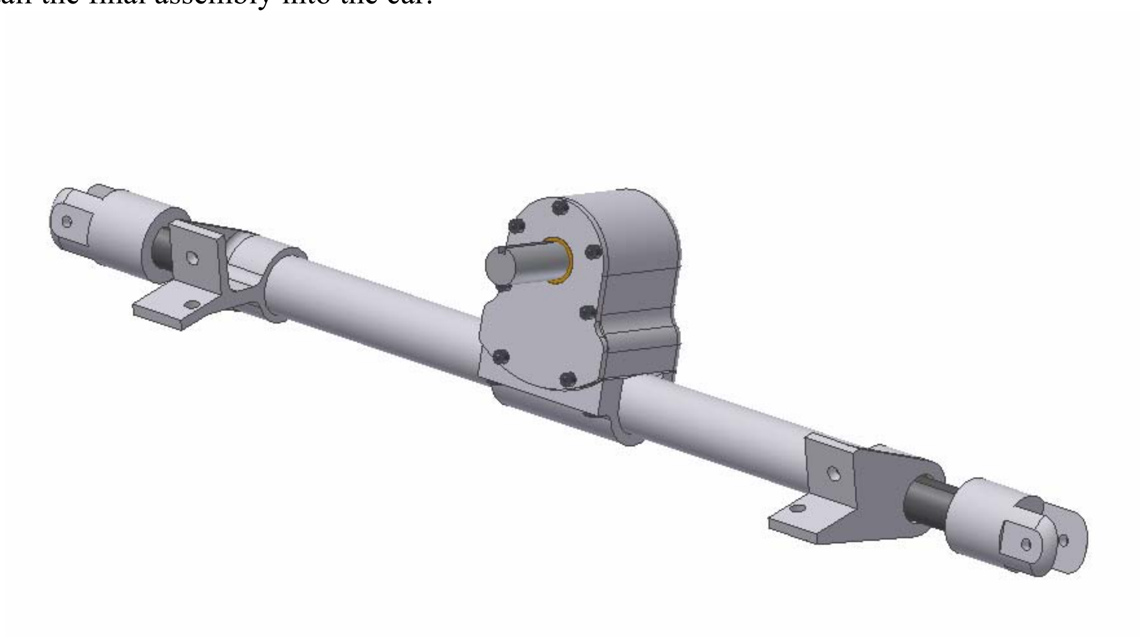


Figure 34. Completely assembled steering rack.

ANALYSIS

All steering rack forces are calculated from the front wheels. Due to suspension geometry, the car will always want to 'unsteer' as it moves through a corner. This is known as the restoring moment and it acts directly on the front tires. The restoring moment can be calculated by summing the moments about the steer axis, which is defined by the suspension geometry.[2] The restoring moment creates stress throughout the steering system and ultimately determines the input force required by the driver. Integrating the steering system with the completed suspension left only the turning conditions to affect the load. The restoring moment was calculated using the equations shown in Appendix 1.

As the car is designed for auto-crossing, the steering system will be subjected to many different turn conditions. Using Microsoft Excel, the restoring moment was calculated depending on lateral, vertical, and tractive loads as well as steer angle, and suspension geometry. See Appendix 2. Provisions for weight transfer under braking and rolling were also included to analyze combined loading. It was found that the vertical loads and braking forces have a negligible affect on the restoring moment. Because the tractive and normal loads contribute less than 1% of the loading, only steer angle and lateral loads are considered in the force analysis. In addition, any unexpected impact loads, such as small bumps along the course, will occur too rapidly for the driver to predict, and therefore only steady state turning conditions were analyzed.

The Bicycle Model of a Car was used to determine the actual turn speeds and radii given a specific steer angle and lateral load.[6] Given the accuracy of the bicycle model, the feasibility of the turn condition was estimated.

Several assumptions were made to simplify the calculations and appropriate factors of safety were applied. These assumptions were minimal but necessary to predict reasonable loads.

Assumptions

All forces are distributed evenly:	Under steady turning conditions, normal and lateral loads will be distributed among front and rear wheels equally. The car is very close to a 50-50 weight distribution, so this is a valid assumption.
Ackerman effects are negligible:	Because the front tires follow different radii throughout a turn, the inside wheel must turn farther to stay online- this is called Ackerman. We assume the angular difference is small and take an average steer angle- the bicycle model of a car.
Smooth Track Conditions:	The autocross courses this vehicle is designed for take place in parking lots with purpose-set tracks. Bumps and sudden impact forces will not be seen under the intended driving conditions.

Regardless of the turn situation, there is a maximum load the tires are able to support. Exceeding this load will cause the car to slide out of control. Based on previous cars, given optimum track conditions, an appropriate limit is 1.5 G. This can be taken as an acceleration limit, braking limit, turning limit, or any combination in which the combined magnitude does not exceed 1.5 G. The combined magnitude is based on the root sum of squares rather than simple addition.

The restoring moment was calculated at several combinations of braking and turning loads, and then resolved through the steering system using the free body diagram in APPENDIX SLUT2. Analyzing the system at maximum lateral force (1.5 G) and maximum steer angle (30 degrees measured at the front wheels) yielded a restoring moment of 495 inch-pounds. This corresponds with a 377 pound lateral force, and a 213 pound longitudinal force acting on the end of the rack gear. A steady turn with a radius of 10 feet maneuvered at 15 miles per hour is required to generate these values. Although, it is unrealistic for the car to maintain traction in such a small radius turn, we will use this as our ultimate load. Maximum lateral acceleration will be achieved near 13 degrees of steer angle. Under this condition, the lateral and longitudinal loads are 292 and 97 pounds, respectively. Designing using the higher loads yields a minimum factor of safety of 1.3 throughout the system.

In a perfectly tuned car, these loads would act evenly on the inside and outside tire, but in reality, this is not the case. The car will experience some degree of roll, which will unload the inside tire while increasing the load on the outside tire. This does not change the required restoring moment, but it does change where the force acts within the steering system. The steering pick-up point is ahead of the steer axis, which means the steering system will 'pull' the outside wheel inward as the car corners. The analysis applies the full load of the restoring moment upon the outside wheel, generating another maximum loading condition. This assumption will prevent failure in the case that the inside wheel lifts off of the ground during a turn. The steering system benefits from the forward mounted pick-up point because buckling and associated deflections will be minimized. The elimination of buckling deflections also eliminates a binding load that would cause the steering to feel 'heavy.' In other words, the steering system will not bind or stick due to normal road loads.

To ensure long lasting components, the teeth on the rack were analyzed for fatigue using AGMA standards and modified Goodman analysis. The gears are machined with high tolerances, ensuring a contact ratio of 1.76. The AGMA bending stress was calculated at the highest point of single tooth contact (HPSTC) and used in the Goodman fatigue calculations. Assuming 100 hours of auto-crossing, 1 minute laps, and 25 turns per lap, the steering rack is expected to undergo 75,000 cycles. As a conservative analysis, each cycle is assumed to be under the maximum load condition. The center teeth will see fully reversed bending, and the outer teeth will see fluctuated bending because the car will 'unsteer' on its own, loading the teeth in only one direction.

MATERIALS SELECTION

The materials used in the steering system target precise operation and light weight components. Although precision and weight are the top priorities, cost, manufacturability, and reliability were also considered.

Precision in the steering system is derived from high manufacturing tolerances and minimal deflection. Deflection in any component leads to steer compliance, resulting in an unresponsive steering system.

The pinion and idler gears were purchased, which limited material options but reduced manufacturing time. Although these gears are steel, they have a smaller face width than a comparable aluminum gear. The rack gear is Turned, Ground and Polished (TGP) 1042 carbon steel rod. The TGP material is more precise than standard round stock and the TGP finish removes surface tension inherent from standard extruding. This reduced the risk of the rack gear bending when the gear teeth were cut. The higher strength of the steel also allowed for extra material to be removed without introducing steer compliance. In addition to the precise material, the steel on steel gear mesh is advantageous for reliability. A softer material in the gear mesh would increase the possibility of galling, and compromise precision over time.

The entire rack housing is made of high strength 6061-T6 aluminum. 7075-T651 aluminum was considered, but the higher grade aluminum was significantly more expensive, and only provided marginal strength increases when compared to 6061-T6 aluminum. The tubular rack housing, pinion housing and rack supports were machined from 6061-T6 aluminum and under maximum expected loads the tubular housing will deflect 0.0003 inches- which is on the same order of magnitude as the suspension arms.

Several components slide against each other but bearings are too heavy and require more housing material. In these situations bearing bronze was used, preventing surface wear with both steel and aluminum components.

FEA ANALYSIS

Finite element analysis (FEA) was performed on many of the critical components in the system. Some of this analysis was performed in the ANYSIS package built into Autodesk Inventor 9 Professional and the rest was performed in the COSMOS package built into SolidWorks 2005. Both of these software packages are very basic solvers that automatically apply a 3D tetrahedral mesh to the part. Aside from the resolution of the mesh, very little parameters can be adjusted with this software. Although this certainly isn't ideal, the results should still be fairly accurate.

The loads applied to components were derived from the analysis described in Appendices 1 and 3. Details of these calculations can be found in the design journal. Material properties were found on www.matweb.com. The factor of safety and deflection characteristics were analyzed for each component under maximum expected loading conditions. Analysis results of some of the critical components are shown below.

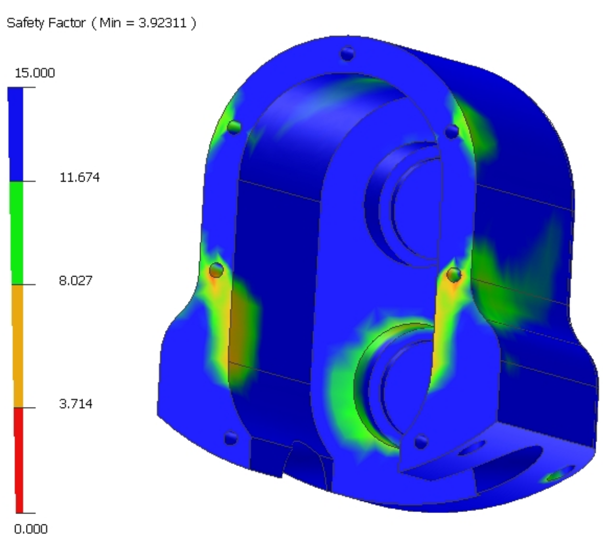


Figure 35. Pinion Housing FOS Results

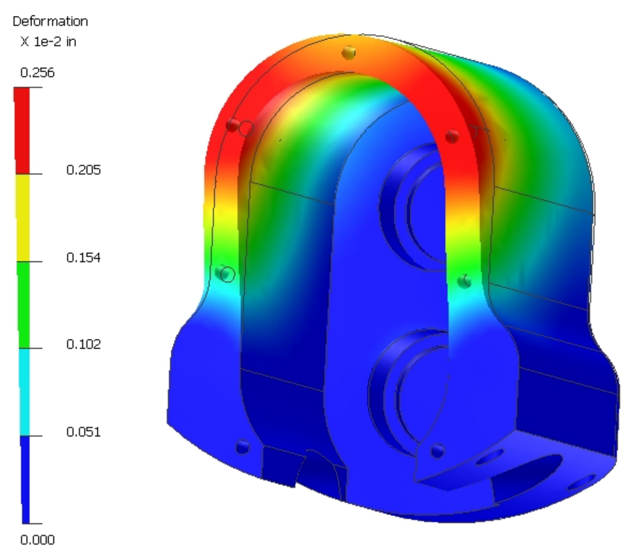


Figure 36. Pinion Housing Deformation Results

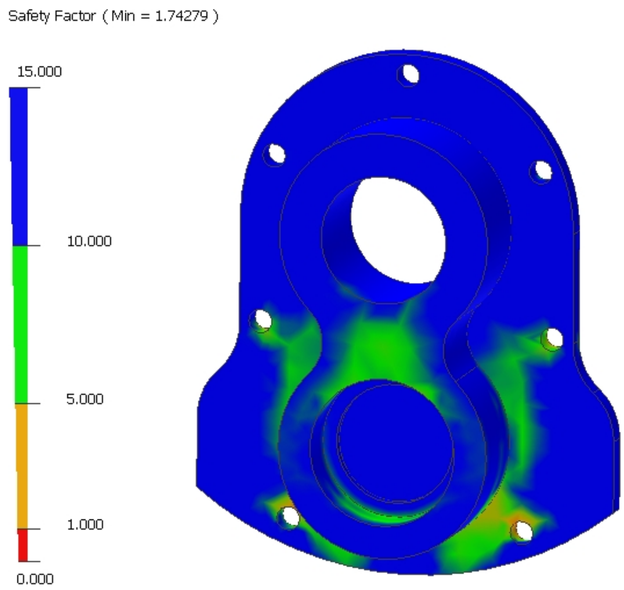


Figure 37. Pinion Housing Cap FOS Results

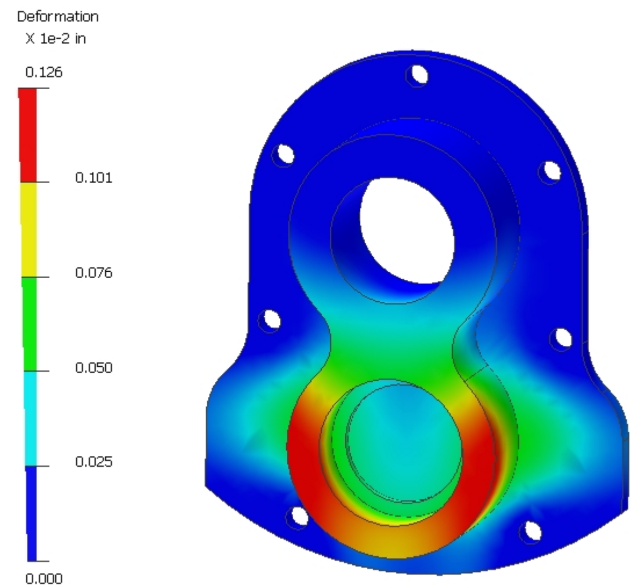


Figure 38. Pinion Housing Cap Deformation Results

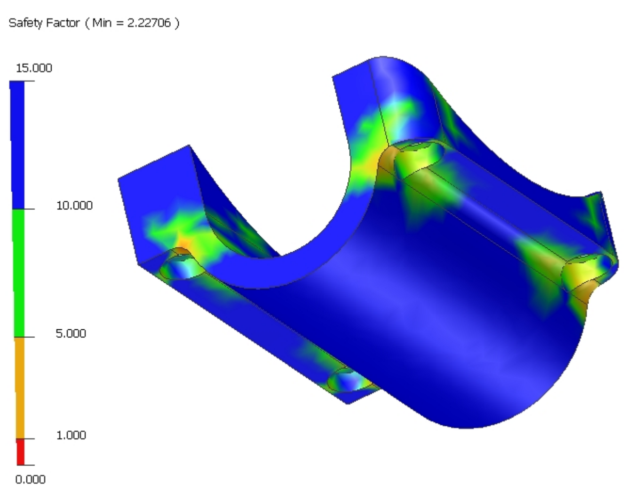


Figure 39. Pinion Housing Clamp FOS Results

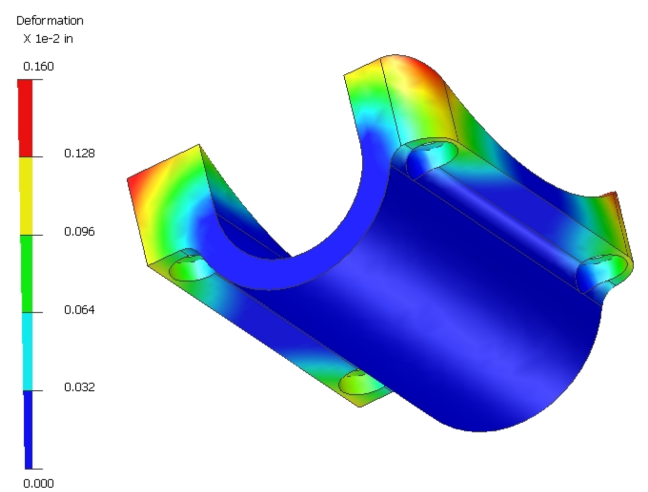


Figure 40. Pinion Housing Clamp Deformation Results

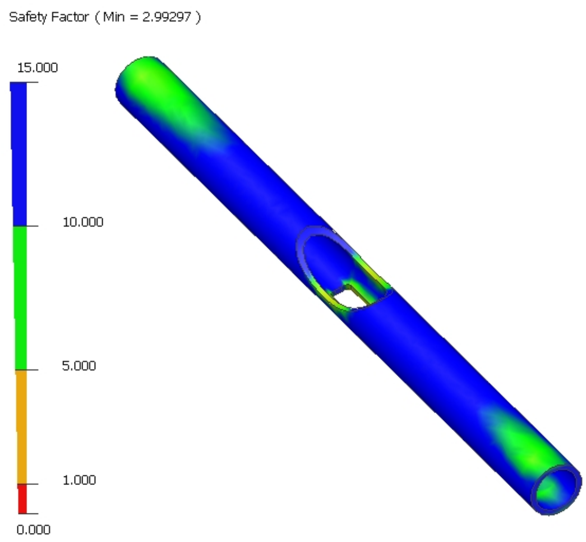


Figure 41. Main Housing Tube FOS Results

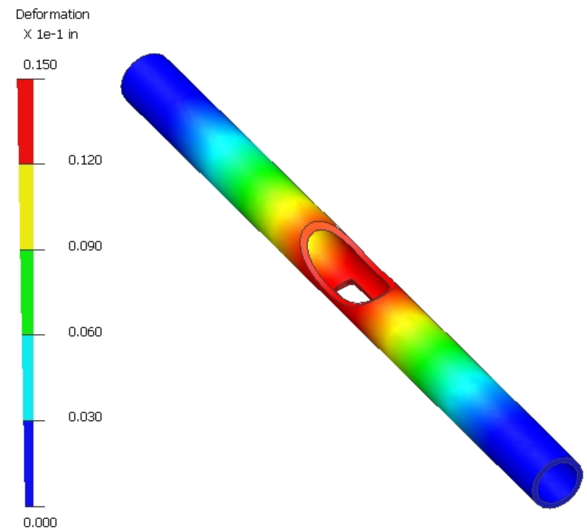


Figure 42. Main Housing Tube Deformation Results

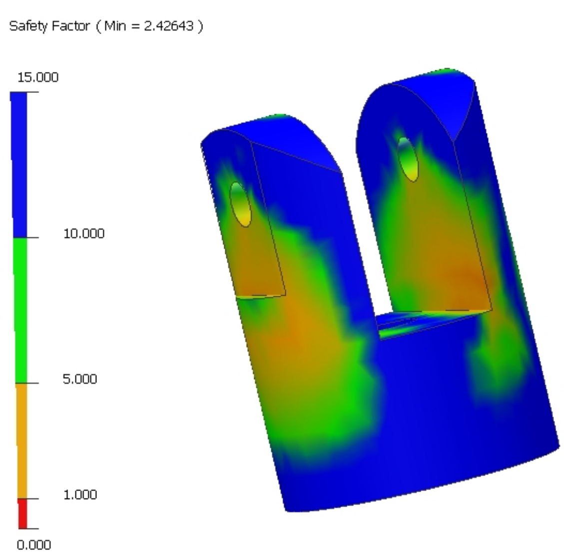


Figure 43. Rack End FOS Results

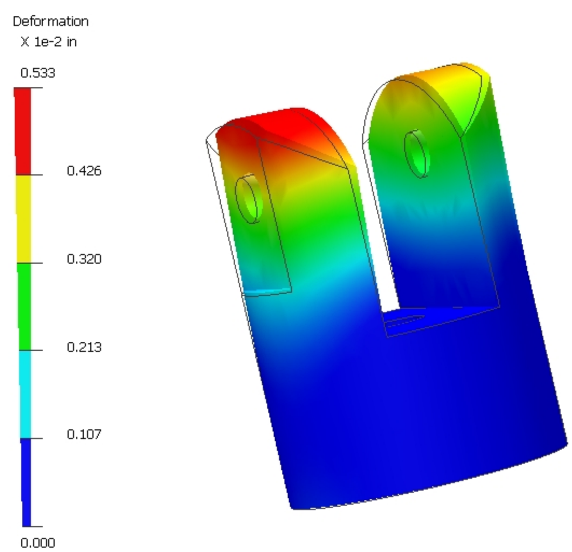
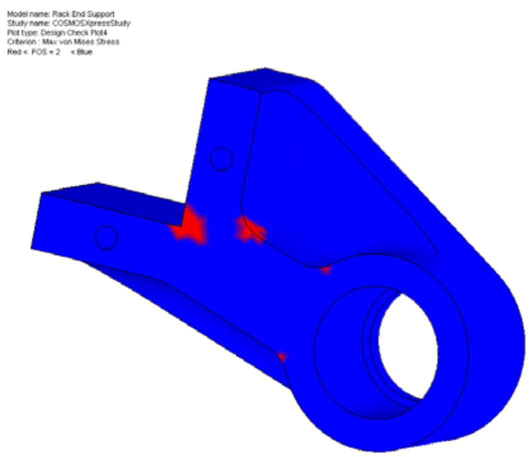


Figure 44. Rack End Deformation Results



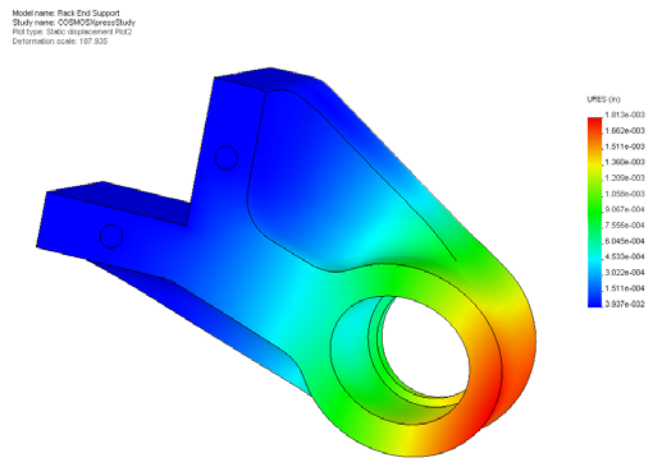


Figure 45. Housing End Support FOS Results

Figure 46. Housing End Deformation Results

PROJECT MANAGEMENT INFORMATION

At the beginning of the project, the overall goals were thoroughly examined. The depth of the project was then considered so that completion was attainable within two quarters. With the direction of the project fully defined, the steps needed to complete it were carefully determined. In organizing a Gantt chart, time was allotted for each phase of the project. As the Gantt chart evolved, the importance of time management was realized.

Preliminary scheduling proved to be fairly accurate as few modifications were made during the progression of the project. Some machining processes were delayed due to uncontrollable factors, but the majority of the manufacturing went faster than expected, a clear indication of careful planning.

The Gantt chart served to be a good motivator and organizer. The changes made, such as shortening the week of testing and revisions did not have many negative effects. The report writing was delayed slightly while work was focused on other aspects of the Formula SAE Car, but this proved not to be critical either. During the quarter as work on the rest of the vehicle increased, our weekly scheduled meetings were replaced by frequent impromptu meetings. Most team members were in the machine shop or the student project design lab so casual discussions covered what we would have talked about in the meetings. The scheduled meetings are left in the revised Gantt chart because it was not practical to record the date and time of every small conversation in passing, but it is still representative of the time we spent communicating as a team.

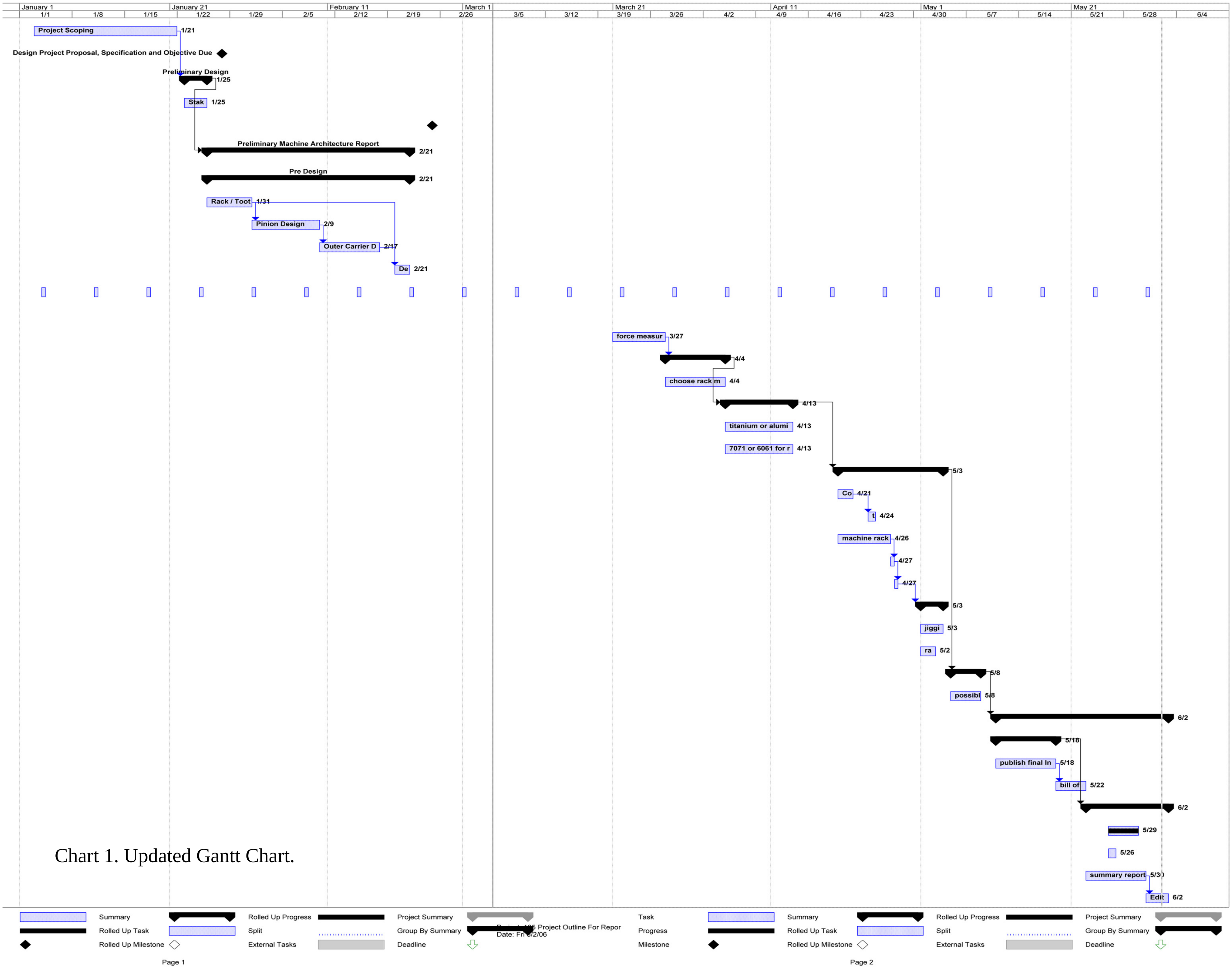


Chart 1. Updated Gantt Chart.

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1. Shigley, Joseph E., Mischke, Charles R., *Mechanical Engineering Design*. Mc Graw-Hill, New York, 6th ed. 2001.

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3. Smith, Carroll, *Engineer to Win*. MBI Publishing Company, Minnesota, 1984.

4. Norton, R. *Machine design*. Pearson Prentice Hall, Upper Saddle River, NJ, 2006.

5. MATWEB Material Property Data; Database of Material Data Sheets. Available at: http://www.matweb.com. Accessed May 03, 2006.

6. Karnopp, D. *Vehicle Stability*. Marcel Dekker, Inc.. New York, 2003.

APPEDIX

APPENDIX 1

Equations

Three factors affect the steering force in our application; vertical load, lateral force and tractive force. Each of these conditions produces a moment acting about the steer axis, and the moment is balanced through the steering rack. The total moment about the steer axis is a summation of the individual moments.

Moment due to Vertical Load:

$$M_V = -(F_{zl} + F_{zr})\, d \sin(\lambda) \sin(\delta) + (F_{zl} - F_{zr})\, d \sin(v) \cos(\delta)$$

Where:

F_{zl}	Vertical Load, Left Wheel
F_{zr}	Vertical Load, Right wheel
d	Lateral Offset at the ground
λ	Lateral Inclination Angle
δ	Steer Angle
v	Caster Angle

Moment due to Lateral Force:

$$M_L = -(F_{yl} + F_{yr})\, r \tan(v)$$

And

$$F_{yl} = F_{yr} = C_f \text{ (mass of car) } /4$$

Where:

C_f	Cornering Force
F_{yl}	Lateral Force, Left Wheel
F_{yr}	Lateral Force, Right Wheel
r	Tire Radius
v	Caster Angle

Moment due to Tractive Force:

$$M_T = (F_{xl} - F_{xr})\, d$$

Where:

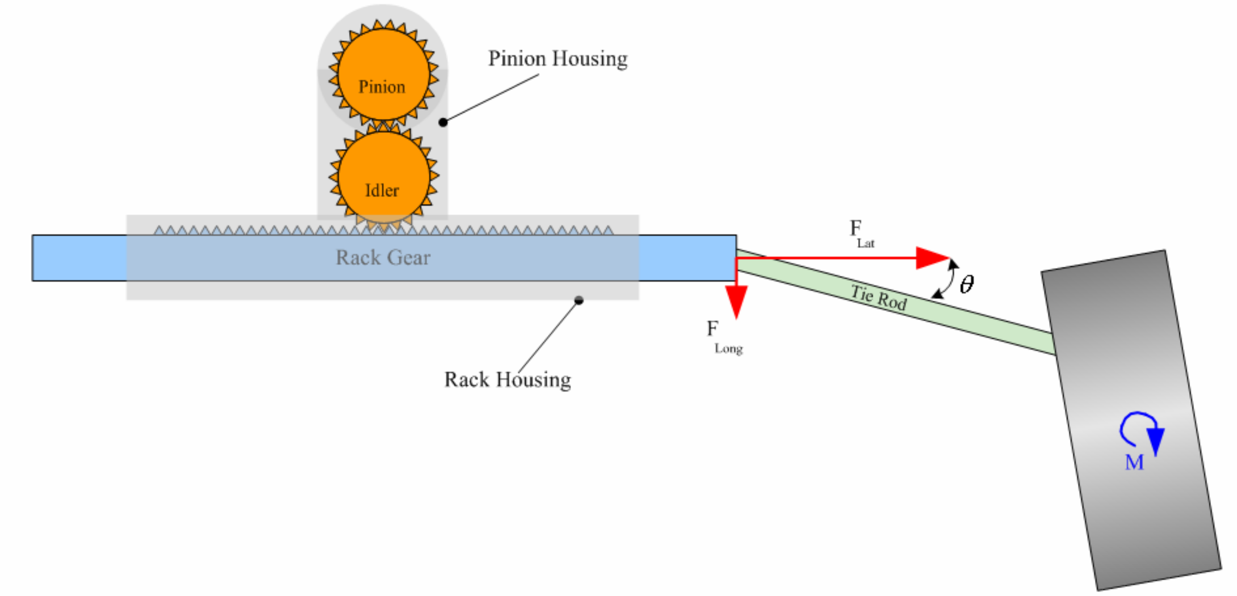
F_{xl}	Tractive Force, Left Wheel
F_{xr}	Tractive Force, Right Wheel
d	Lateral Offset at the Ground

APPENDIX 2

Free bodies

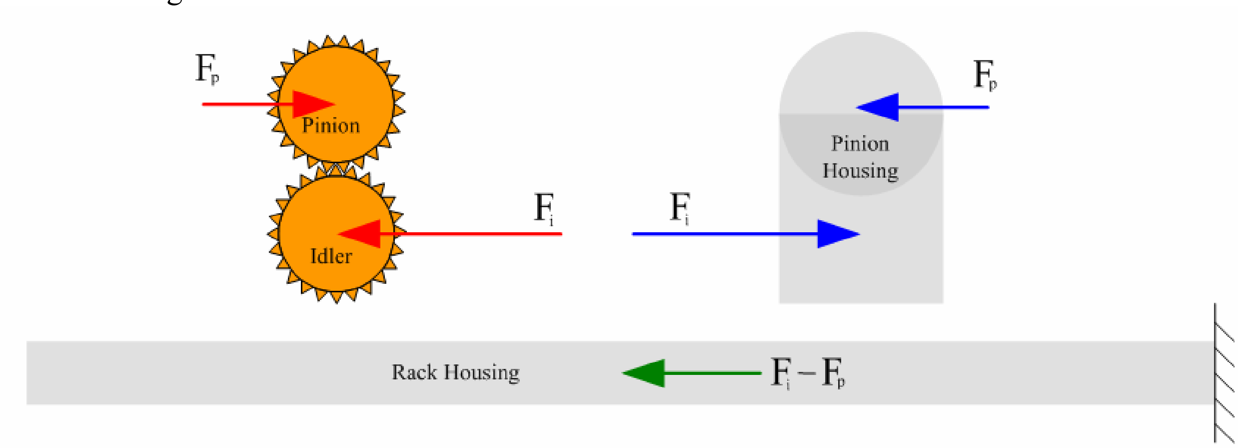
The following free body diagrams both show the right side (outside) of the steering system in a left hand turn.

Restoring moment



This shows the restoring moment acting from the outside wheel to the steering rack. The angle, θ, changes as a function of steering angle. These are the longitudinal and lateral forces used to calculate rack deflection, tooth fatigue and housing stress. These are applied forces.

Rack Housing



The components acting from the rack gear to the pinion housing was determined using this free body. For simplicity, the longitudinal components are removed, as the rack supports will counter that load. All forces shown are resultant forces, and it should be noted that if the rack gear is in tension, the housing is in compression.

Appendix 3

Shown on the following pages are the values of inputs and resulting outputs for a steady turn taken at our maximum loading. The steering input is 30 degrees and the cornering acceleration is 1.5G.

Suspension/Cornering Inputs			
Mass of the Car	m	660	lbs
Lateral Offset at the ground	d	0.3	in
Lateral Inclination Angle	λ	11.53	degrees
Steer Angle	δ	30.00	degrees
Caster Angle	v	5.7	degrees
Cornering Accel	C_f	1.50	g's
Braking Load	F_B	0.00	g's
Tire Radius	r	10	in
Vertical Load, Left Wheel	F_{zl}	165.0	lbs
Vertical Load, Right wheel	F_{zr}	165.0	lbs

Housing/Rack Dimensions		
Old Housing Width	18.3	in
New Housing Width	14	in
Eye-Eye length	22	in
Face Width	0.5339	in
Dimetral Pitch	16	in
Pinion Pitch Radius	0.625	in

Material Properties			
Steel TGP 1045			
Rack Diameter	D	0.75	inches
Bore Diameter	d	0.5	inches
Wall Thickness	t	0.125	inches
Moment of Inertia	I_y	0.012	in^4
Area	A	0.442	in^2
Yeild Strength	S_Y	58700	psi
Ultimate Strength	S_U	97900	psi
Fatigue Strength (N=5E8)	S_f'	48950	psi
Young's Modulous	E	10400000	psi
Shear Strength	τ_{xy}	48000	psi

Turn Conditions		
Radius	10	ft
Velocity	15	mph

Force Components, lbs	
Lateral	377
Longitudinal	213

Rack Tooth Forces lbs	
W	401
W_t	377
W_r	137

Longitudinal Deflection			
Max Moment	**M**	1045	lb-in
Bending Stress	**σ**	31429	psi
Longitudinal Deflection	y_{max}	**0.03231**	in
Factor of Saftey	**η**	3.1	*Failure
	η	1.9	*Yield

Axial Deflection			
Compressive Force	**F**	377	lb
Axial Stress	**σ**	854	psi
Later Deformation	**Δs**	**0.0009773**	in
Factor of Safety	**η**	115	*Failure
	η	69	*Yield

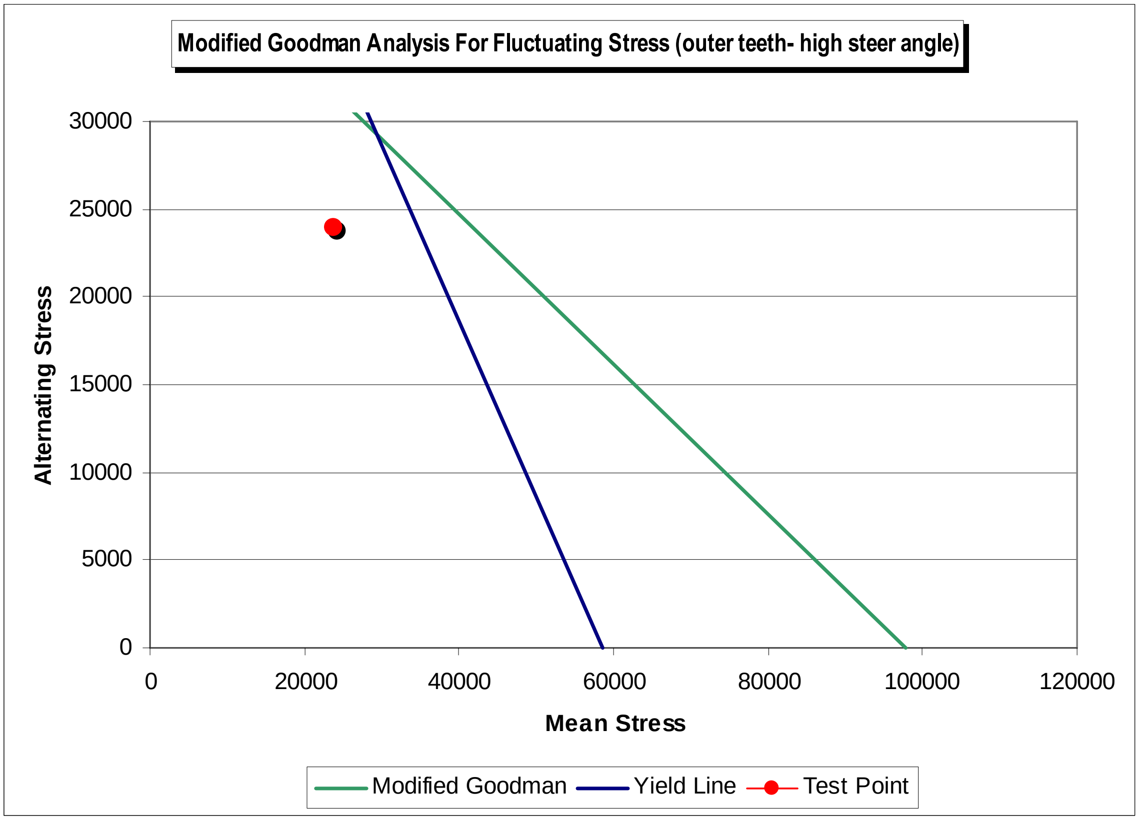
Driver Input		
Steering Wheel Radius	4.75	in
Force Required	**50**	lbs

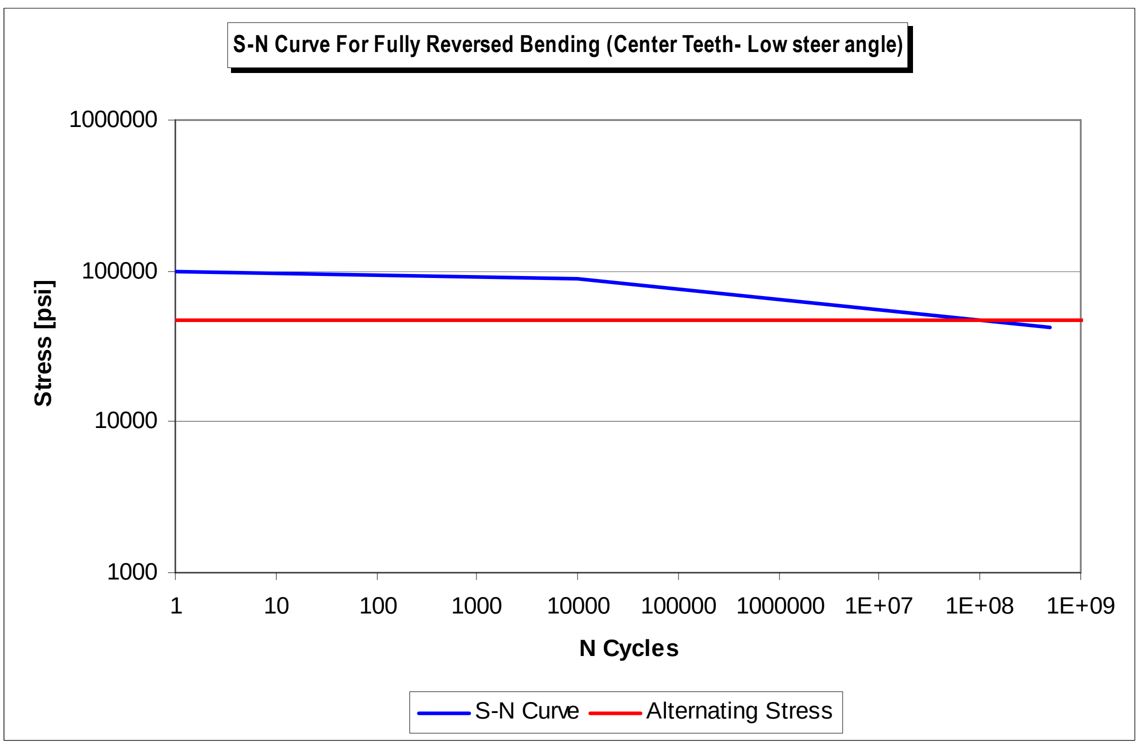
Fatigue Failure* (rotating beam theory)			
Adjusted Fatigue Strength	S_f	41852	psi
Fully Reversed			
Mean Stress	σ_m	0	psi
Alternating Stress	σ_a	47779	psi
Fluctuating			
Mean Stress	σ_m	23889	psi
Alternating Stress	σ_a	23889	psi
Scaling Factors			
Load	C_{load}		1
Size	C_{size}		1
Surface Finish	C_{surf}		0.95
Operating Temperature	C_{temp}		1
Required Reliability	$C_{reliab.}$		0.9

AGMA Bending on Rack Teeth				
Bending Stress	σ	47779	psi	
Factor of Saftey	η	2.0	*Failure	
Scaling Factors				
Geometry		J	0.43	*assuming higher presicion-> load sharing
Dynamic		K_a	1	*incredibly low pitch-line velocity
Load Distribution		K_m	1.6	*high face-width factor (F/P_d), but still relatively narrow face (<2in.)
Application		K_v	1.25	*assumes light shock on rack, uniform load application on pinion, could be 1
Size		K_s	1	*no real set of values… we have small teeth
Rim Thickness		K_B	1	*used for large gears, with spokes (sprocket)
Idler		K_I	1.42	*set to 1.42 if testing idler gear

Housing Tube Deflections			
Tube Diameter	D	1	inches
Bore Diameter	d	0.095	inches
Wall Thickness	t	0.4525	inches
Moment of Inertia	I_y	0.049	in^4
Area	A	0.785	in^2
		0.0003232	inches

Steering Column Deflections			
Column Diameter	D	0.625	inches
Bore Diameter	d	0.495	inches
Wall Thickness	t	0.065	inches
Moment of Inertia	J	0.009086	in^4
length	l	19	inches
		2.435	degrees


Modified Goodman Analysis For Fluctuating Stress (outer teeth- high steer angle)
Alternating Stress
30000
25000
20000
15000
10000
5000
0
0
20000
40000
60000
80000
100000
120000
Mean Stress
Modified Goodman
Yield Line
Test Point


S-N Curve For Fully Reversed Bending (Center Teeth- Low steer angle)
Stress [psi]
1000000
100000
10000
1000
1
10
100
1000
10000
100000
1000000
1E+07
1E+08
1E+09
N Cycles
S-N Curve
Alternating Stress