

Design of an Isolated, Quiet, Universal ATX Computer Case

by

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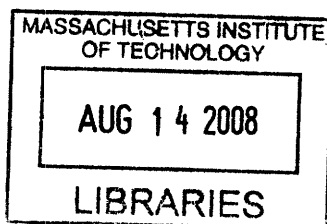
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Abstract

Machine integrators and PC builders require a great deal of flexibility and durability when assembling electronic components. The aim of this project was to design a universal ATX computer case that could accommodate any motherboard and processor combination available on the market. It also had to be isolated from the outside air, as to be used in dirty or industrial applications where durability is required. This case design maintains low cost and high flexibility by dissipating high amounts of heat using natural convection on the outside of the case and forced convection on the inside of the case. It is designed to dissipate a total of 250 W through a large heat exchanger and 27 W through a smaller one dedicated to dissipating heat from the hard disk. A universal CPU cooler uses pin fins with forced convection and custom adapter plates to mate with a variety of CPU processors. It is capable of dissipating 162 W, more than any CPU on the market needs. This is all accomplished without exceeding any surface temperature limit of any of the major electronic components in the case can operate in ambient air temperatures of 40 C.

Thesis Supervisor: John H Lienhard
Title: Professor of Mechanical Engineering

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Chapter 1: Introduction

Currently computers designed for a specific task dominate the industrial computing market. Due to the requirement that computers in industrial settings have to be isolated from a dirty environment custom-built machines are required for a given task. These machines are custom designed from the outside cases, to the shape and size of the electronics inside them. They however use standard processor chips, and standard operating systems, like Microsoft Windows. The only thing preventing such a machine from being replaced by a typical office desktop PC that uses a standard motherboard and interface cards is that standard ATX form factor PCs cannot stay cool without circulating air from the outside through them. In an office environment this may be acceptable since levels of dust and damaging debris are low, and the internal components are not damaged. However this is not the case in an industrial environment where airborne debris may damage the components. The aim of this project was to design a universal ATX computer case that can keep the internal electronics cool without exchanging air with the outside, while being cost effective (namely, not use water cooling). This case would also have the added benefit of being “silent” by industry standards to attempt to capture some of the silent PC user market. This case would increase the flexibility of industrial machine integrators when designing industrial automation systems.

Chapter 2: Concept Selection

2.1 Functional Requirements and Design Parameters

Given the objective of the design several functional requirements were established.

1. Internal components must be isolated from ambient air.
2. Case must be “silent,” or emit less than 30dB of noise.
3. The case must accept *any* ATX motherboard.
4. The assembled computer can operate in ambient air temperatures up to 40 C.

In addition to these functional requirements ease of manufacturability and low cost were also added as design requirements.

The first functional requirement required that main heat dissipation out of the case be done by convection. This required the use of high conductivity metal, like 6061 Aluminum. To aid internal convection, fan could be used, but there could be no open holes in the case that would allow air to pass through.

The fact that the case needed to be “silent” required a very quiet slow speed fan, or no fan, be used to aid convection on the outside. The fact that the case is sealed makes using faster fans on the inside feasible. Because of this functional requirement the CPU could not be cooled using standard high-speed fans and heat sinks, as these fans are very loud. Therefore a custom CPU heat sink had to be designed so heat could be dissipated using large low-speed fans. This heat sink had to be adaptable to many different types of CPUs as well.

The third functional requirement dictated the overall size of the case, which required a

space for the motherboard of at least 304.8 mm by 243.84 mm (12 in x 9.6 in)¹. The case also had to be able to deal with the average heat dissipation of all the motherboard components.

The last functional requirement would influence the design of the case heat exchanger, which would carry the heat load of all the internal components.

2.2 Strategy Selection

Several strategies were considered using the functional requirement and design parameters as guidelines.

They are as follows...

1. Natural convection on the outside with limited forced Convection inside (single circulating fan), no dedicated processor fan, and expansion slots
2. Same as 1, without expansion slots, which gives a slimmer case
3. Forced convection inside and outside, without a dedicated processor fan
4. Same as 1, with a dedicated processor fan
5. Same as 3 with a dedicated processor fan
6. Natural Convection both inside and outside without expansion slots

All strategies employed an external power supply with dedicated hard disk cooling.

Keeping in mind the functional requirements and what would best satisfy the intended user of the case the following Pugh chart was used to pick the best strategy. The scale was from -3 to +3 with the first strategy being used a baseline. Ratings were derived from how much better or worse the subsequent strategies were at meeting the functional requirements compared to the first one.

¹ Intel Corporation. ATX Specification v2.2. 2003.
<http://www.formfactors.org/developer%5Cspecs%5Catx2_2.pdf> (15 May 2007)

FR→ Strategy	1	2	3	4	TOTALS
1	0	0	0	0	0
2	0	0	-1	0	-1
3	-2	-2	0	+2	-2
4	0	-1	+1	+1	+1
5	-2	-2	+1	+3	0
6	0	+1	0	-2	-1

Table 1: Strategy Selection Pugh Chart

In the end strategies 1 and 6 were chosen, as strategy 4 was very similar to 1. The one fundamental difference between the two was the use of natural convection and the inclusion of expansion slots. The inclusion of expansion slots and the use of forced convection in the case would make heat management easier and thus make the design more flexible, or more “universal,” which is the most important functional requirement.

2.3 Concepts and Final Selection

Three concepts were investigated, and because of similarities these three were condensed into two.

Both concepts had the following common elements, dictated by the functional requirements.

1. Power supply external to the case.
2. Forced convection on the inside of the case.
3. Custom CPU heat sink.
4. Flow director to promote even convection of heat on the inside

The first concept used a smaller case with no expansion cards. It utilized natural convection on the outside, and a baffle in the bottom center of the case to direct uniform

airflow upward into the electronics, taking in hot air at the top of the case at the intake of the primary case heat exchanger. This design used a large passive processor heatsink that would be cooled by airflow from main internal case fans. The CD drive and hard disk drive would be installed on the top or front of the case to take advantage of direct heat dissipation to the environment, reducing the total heat load on the main heat exchanger.

The second concept allowed the use of expansion cards to increase flexibility for the user. It utilized a baffle on the top, or front of the case, depending on its final orientation, to direct air into the primary case heat exchanger forcing air out into the bottom. The hard disk and CD drive would also have independent heat dissipation. It used a fan on the processor and could be installed in any orientation so its flow direction would match that of the primary circulating fans. The two concepts were again compared with a Pugh chart, shown in Table 2. Concept one was used a baseline with Concept 2 compared to it on the same -3 to +3 scale as the strategy selection.

FR → Concept	1	2	3	4
1	0	0	0	0
2	0	-1	+2	0

Table 2: Concept Selection Pugh Chart

In the end Concept 2 was selected, due to superior universality (Functional Requirement 3), despite the fact that it was slightly noisier. Considering that the primary users would be industrial machine integrators, who would care less about noise and more about flexibility, concept 2 would be a better fit.

2.3 Bounding the Problem

In order to establish quantitative specifications for the case, the heat dissipation rate and design limits of several key components in a computer needed to be established. The key heat-generating components of a PC are the power supply (if internal to the case), the CPU, the hard drive, and the graphics card. The greatest of these sources is the CPU, which is usually cooled by a heat sink and high-velocity fan. To establish an upper limit

for heat dissipated by a CPU, data was gathered from the major manufacturers of PC processors, Intel and AMD. This data is shown in Table 3.

Processor	Dissipation req'd [W]	Max Case Temp [C]
Intel® Core™2 Quad Processor Q6600	105	62.2
Intel® Core™2 Duo Desktop Processor E6700	65	60.1
Intel® Pentium® 4 Processor Extreme Edition 3.73 GHz	115	72.8
Intel® Pentium® D processor 960	130	68.8
Intel® Celeron® Processor 2.80 GHz	68.4	75
Intel® Celeron® D Processor 365	65	64.4
AMD Athlon™ 64 Processor	89	70
AMD Athlon™ 64 FX Processor	89	70
AMD Sempron™ Processor	62	70
AMD Athlon™ 64 X2 Dual Core Processor	110	65

Table 3: Heat Dissipation and Temperature Limits for common PC processors.²

From this research and upper limit of 130 W of heat dissipation was established, where the minimum case temperature needed to be below 65 C. This would govern the design of the processor heat sink and substantially effect how much heat the case would have to dissipate.

Hard drives also contribute substantial heat to a system. Due to the similarity of design for standard PC hard drives, the heat dissipated is governed by the speed at which the disk spins, either 7200 RPM or 10,000 RPM. Representative heat dissipation numbers were gathered for hard drives made by Western Digital. This data is found in Table 4.

² Intel Corporation. Processor Specification sheets, 2007.

<<http://processorfinder.intel.com/details.aspx?sSpec=>> (12 February 2007)

AMD. White Papers and Technical Documents., 2007. <http://www.amd.com/us-en/assets/content_type/white_papers_and_tech_docs/30430.pdf> (12 February 2007)

Hard Drive Type	Dissipation [W]			
	Idle	Startup	Seek	Read/Write
WD Caviar RE2 WD4000YR (7200 RPM spin speed)	7.09	22.42	11.43	8.37
WD Raptor WD740GD (10K RPM spin speed)	8.04	27.15	11.42	9.57/9.86

Table 4: Heat Dissipation For A Standard PC Hard Disk³

The hard drive heat sink was therefore designed to manage 27 W of power dissipation, with only a modest temperature increase across the heat sink. This however would not last long as hard drive startup is brief and not repetitive.

Heat dissipation from the power supply was managed by making the power supply external to the case. This is a common practice for small form factor computers, like the Apple Computer Mac Mini. This removes a substantial heat load from the case, and therefore does not need to be considered in the calculations.

The last component is the graphics card. Graphics cards range from those built into the motherboard, to large high-performance cards that can dissipate as much heat as the primary CPU. Using the total power consumption of ATI graphics cards as a benchmark, the largest of which consumed 63 W⁴, a maximum dissipation number of 60 W was established. This notes the fact that no graphics card can dissipate more power than it consumes, due to conservation of energy. While this limit may seem low when compared to high end graphics cards of today, one must keep in mind that this case is not intended for the high end PC gamer market, which required powerful graphics cards. No custom heat sink was designed as all graphics cards have ways of dissipating heat into the case air stream provided by the manufacturer.

³ A. Karabuto. HDD Diet 2: Power Consumption and Heat Dissipation of Enterprise Hard Disk Drives. March 2, 2006. <<http://www.digit-life.com/articles2/storage/hddpower-pro.html>> (12 February 2007)

⁴ T. Tschelblockov. Power Consumption of Contemporary Graphics Accelerators Part I: Graphics Cards on ATI Chips, July 27, 2004. <<http://www.xbitlabs.com/articles/video/display/ati-powercons.html>> (12 November 2007)

Taking into account all these heat source in a typical computer the number of 250 W of total dissipation required by the case was established. This took into account the CPU at 130 W, the graphics card at 60 W and other electronic components that would dissipate heat into the circulated air stream of the case, which were about 10 W. This was added together and multiplied by a safety factor of 1.25 to obtain 250 W of total heat dissipation required.

Chapter 3: Thermal Design

3.1 Overall Resistance Model

Since heat would be dissipated from the case in steady state an overall thermal resistance model was developed. Figure 1 shows a map of resistors and principle heat loads from the case, as described in the previous chapter.

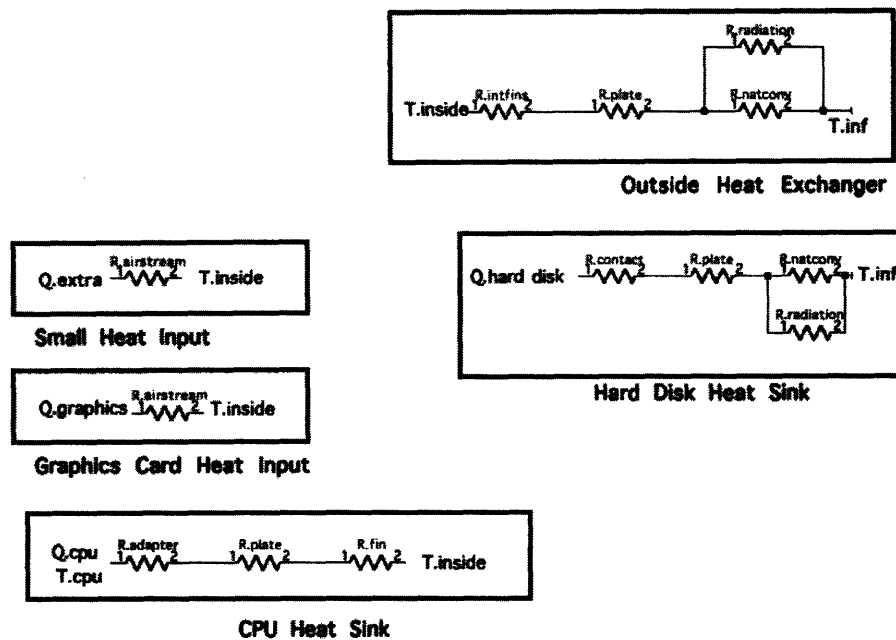


Figure 1: Resistance Network

The three boxes on the left represent heat that is routed through the outside heat exchanger, where the hard disk heat is dissipated separately through its own heat sink, which is diagramed on the bottom right. Complete explanations of these diagrams will be in sub-chapters to follow.

3.2 Outside Heat Exchanger

The outside heat exchangers are responsible for dissipating most of the heat from the case. They rely on forced convection on the inside and natural convection on the outside. Figure 2 shows one half of the outside heat exchanger.

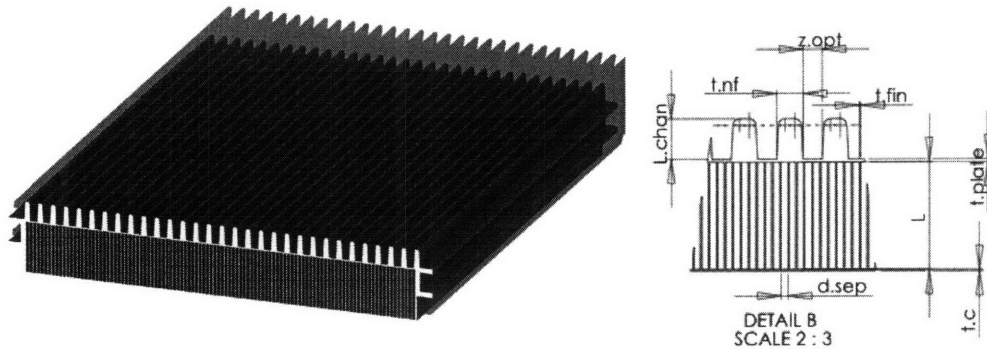


Figure 2: Outside Heat Exchanger with Critical Dimensions (Right)

Several sample calculations were made for the outside heat exchanger. The first design was a simple flat plate for the outside and wide channel on the inside. This channel was 4 cm deep, 30 cm wide and 40 cm tall. The outside plate was simply a flat piece of aluminum 30 cm side and 40 cm tall, and 1 mm thick. In this configuration the total resistance was 0.756 K/W, which was unacceptably high, seeing that this part had to dissipate 250 W.

The next logical step was to add fins to the outside to aid the natural convection, and to the inner channel to reduce the forced convection resistance.

An arbitrary fin thickness of 0.5 mm was picked with a spacing of 1 mm between each fin. This gave a total of 150 fins on each side of the heat exchanger. A hydraulic diameter was calculated and a Reynolds number was determined, basing the velocity as air being driven through 2 standard 120 mm computer case fans. The flow through the channel between two fins, was determined to be laminar, with a Reynolds number of 25, which is not conducive to heat transfer. Even so a Nusselt number and heat transfer coefficient, h , was calculated. The heat transfer coefficient was determined to be $61.5 \text{ W/cm}^2 \text{ K}$. These

were calculated using Equation 1, which are basic figures used in calculating convection coefficients.

$$\text{Re} = \frac{\rho V d_h}{\mu} \quad (1a) \quad \text{Nu} = 8.235 \quad (1b) \quad h = \frac{\text{Nu} k}{d_h} \quad (1c)$$

where V is the stream velocity, μ is the viscosity of air, Nu is the Nusselt number, d_h is the hydraulic diameter or 4 times the cross sectional area of a channel divided by the perimeter, and k is the conductivity of air. 8.235 is the Nusselt number for laminar flow between two parallel plates with constant heat flux⁵, which is the case with the small channels between the fins.

The quick set of calculations above proved that forced convection on the inside has the capability to dissipate the heat required. These values were changed were changed later during optimization

In order to be able to transfer enough heat an array of fins was used on the inside. The resistance of the fin array on the inside was calculated using equations from *Extended Surface Heat Transfer* by Aziz, Kraus, and Weltley⁶. Two m-factors one for the fins, noted by the subscript f, and one for the cover plate, noted by subscript c. are defined in equations 2a and 2b.

$$m_f = \sqrt{\frac{2h}{kt}} \quad (2a) \quad m_c = \sqrt{\frac{h}{kt_c}} \quad (2b)$$

where h is the heat transfer coefficient for a individual fin, k is the conductivity of the metal, and t the thickness of the fin, and t_c is the thickness of the coverplate.

⁵ J.H. Lienhard IV, and J.H. Lienhard V, A Heat Transfer Textbook 3rd ed. (Phlogiston, Cambridge, MA, 2006), p.374

⁶ A.D. Kraus, A. Aziz, and J. Welty, Extended Surface Heat Transfer 1st ed. (Wiley, New York, 2001), p.396-397

These factors are used in two fin admittances described by equations 3a and 3b

$$Y_{of} = \sqrt{2hkt}W \quad (3a) \quad Y_{oc} = \sqrt{hkt_c}W \quad (3b)$$

where Y stands for the individual fin admittances, and W is the length of the base of the fin; the dimension perpendicular to the thickness.

These admittances were plugged into an equation that gives the overall fin admittance, which is described by Equation 4.

$$Y_{in} = \frac{\tanh(m_f L) + 2 \left(\frac{Y_{oc}}{Y_{of}} \right) \tanh(m_c d_{sep})}{1 + 2 \left(\frac{Y_{oc}}{Y_{of}} \right) \tanh(m_c d_{sep}) \tanh(m_f L)} \quad (4)$$

where L is the end to end length of the fins in the array, and d_{sep} is the separation distance between each fin in the array. In these equations the cover plate is assumed to be adiabatic on one side. This fits with the heat exchanger design as the tip of the fins, closest to the cover plate, would be approximately the internal air temperature of the case, which would also be the temperature on the adiabatic side of the cover plate. Therefore no heat should move across the cover plate directly. Even if heat does move across the coverplate, it would only decrease the thermal resistance, making the actual situation better than the model.

The outside fins on the heat exchanger had to rely exclusively on natural convection. Because natural convection is dependent on the surface temperature of the fin, a temperature had to be guessed. In order to obtain the best heat dissipation on the outside, a optimum spacing was found using Equation 5.

$$z_{opt} = 2.71 \left(\frac{g \left(\frac{1}{T_{inf}} \right) (T_s - T_{inf})}{\alpha \nu L} \right)^{-1/4} \quad (5)$$

where g is the acceleration of gravity, T_{inf} is the ambient air temperature, T_s is the estimated fin surface temperature, L is the end to end length of the fin, α is the thermal diffusivity of air, and ν is the kinematic viscosity of air. The recommended optimum spacing gives the most effective fin pattern to dissipate the most heat.

Next a Rayleigh number based on constant heat flux was calculated, given by Equation 6.

$$Ra = \frac{g \left(\frac{1}{T_{inf}} \right) z_{opt}^4}{\alpha \nu k} \frac{Q_{tot}}{2A_s} \quad (6)$$

where k is the conductivity of fin material, in this case aluminum, Q_{tot} is the total heat dissipated by the case in Watts, and A_s is the surface area of the plate on one side of the outside heat exchanger. This number is not dependent on the surface temperature of the fins, and is therefore a more accurate.

A Nusselt number, based on the spacing, z , would give the heat transfer coefficient for the fin. The Nusselt number is also for constant heat transfer and is given by Equation 7.

$$Nu_z = 0.144 \sqrt{Ra \frac{z}{L}} \quad (7)$$

A heat transfer coefficient was calculated from the Nusselt number based on Equation 1c. Fin resistance and heat exchanger correlations were taken from the 2.006 course notes by Brisson, et al⁷. Two factors were calculated, given by Equation 8.

$$m = \sqrt{\frac{hP}{kA_{cx}}} \quad (8a) \quad M = \sqrt{hPkA_{cx}} \quad (8b)$$

where h is the heat transfer coefficient calculated above, k is the conductivity of the fin material, P is the wetted perimeter of the fin, and A_{cx} is the cross-sectional area of the fin.

The fin admittance was then calculated using the formula given in Equation 9. This formula assumes the tip of the fin is adiabatic.

$$Y = M \frac{\sinh(mL) + \frac{h}{mk} \cosh(mL)}{\cosh(mL) + \frac{h}{mk} \sinh(mL)} \quad (9)$$

where M and m are the two factors calculated in Equation 8, and Y is the fin admittance.

The radiation component of the outside fins of the heat exchanger would be very complicated to explicitly calculate, and it was therefore assumed that it would increase the admittance by a factor of 2.

All the fin admittances need to be combined into a single admittance factor, or UA for the heat exchanger. The outlet temperature and maximum dissipated heat were calculated using standard heat exchanger equations. They are given in Equations 10 and 11.

$$T_{out} = (T_{in} - T_{inf}) \exp\left(\frac{-UA}{\dot{m}c_p}\right) + T_{inf} \quad (10)$$

⁷ J.G. Brisson, et. al. 2.006 Course Notes, Massachusetts Institute of Technology, 2006.

$$Q_{\max} = (T_{in} - T_{out})UA \quad (11)$$

where T_{out} is the outlet temperature of the heat exchanger, T_{inf} is outside air temperature, T_{in} is the inlet temperature, which is determined by the maximum output of the CPU heatsink plus a safety factor, UA is the total admittance calculated earlier, $m.\dot{}$ is the mass flow rate of air from the fans, and c_p is the heat capacity at constant pressure, for air.

3.2.1 Design and Optimization

The main case heat exchanger was designed to optimize natural convection on the outside and forced convection heat transfer on the inside. The fins on the outside also had to be unobtrusive and avoid injuring someone should they bump into the case. Figure 3 shows a complete heat exchanger in its installed orientation.

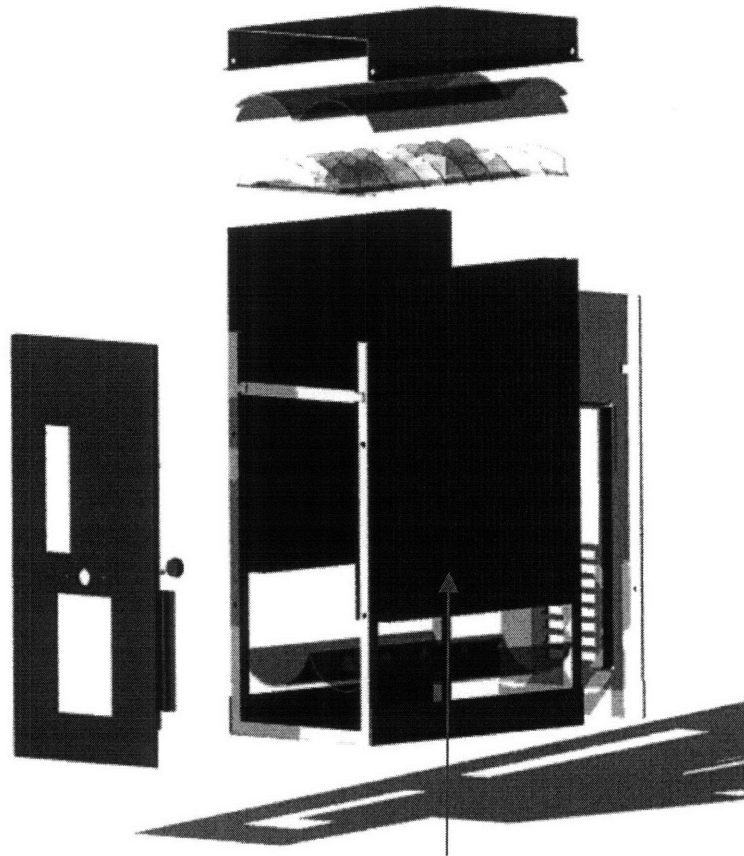


Figure 3: Main Case Heat Exchanger (indicated by arrow)

The outside fins were wide and had rounded corners to eliminate sharp edges. The fins were also angled inward as they got longer, to save material and reduce weight. This is because a smaller cross sectional area is required to conduct heat as the fin gets longer, because heat is removed by convection from the outside of the fin. This principle follows the principle of optimum fin theory.⁸

To optimize the design of the fins the value of mL was maximized given other design constraints. m is the extinction length of the fin in units of $1/m$, when multiplied by L it gives a dimensionless value that when it is the argument of the $\tanh()$ function give the fraction of the total possible admittance that a fin can achieve. An mL value of 1.5 means the fin can dissipate 90% of the maximum amount of heat that a longer fin of the same

⁸ J.H. Lienhard. 2.51 Lecture Notes, Massachusetts Institute of Technology, 2007

shape can dissipate. The outside fins were made rather short, constrained by the need to minimize weight and maintain an aesthetically pleasing, safe exterior. They achieved an mL of 0.52. While this is a great deal less than 1.5, it has to be considered that radiation dissipates as much heat as natural convection in this case, and can make up for the convection loss in the fin. The outside fins were also made black to maximize the dissipation from radiation.

Despite these apparent shortcomings the main heat exchanger achieved excellent heat dissipation. Table 5 describes the dimensions chosen, and the resulting heat transfer characteristics.

Parameter	Inside Fins	Outside Fins
Fan Flow Rate	56 CFM	N/A
Fin Length	0.04 m	0.015 m
Cross Sectional Area	0.0002 m ²	0.006 m ²
Fin Thickness	0.0005 m	0.005 m
Number of Fins	100	28
Conductivity	204 W/(m*K)	204 W/(m*K)
Baseplate Dimensions	0.30 m x 0.40 m	0.30 m x 0.40 m
Baseplate Thickness	0.0015 m	0.0015 m
Separation Distance (b/t Fins)	0.003 m	0.0055 m
Heat Transfer Coefficient, h	40.6 W/(m ² *K)	22.1 W/(m ² *K)
Total Admittance (for Array)	83.8 W/K	14.3 W/K
m*L	1.128	0.52

Table 5: Heat Exchanger Design Parameters

Taking the total admittance of the array and multiplying it by 2 to factor in radiation. The complete heat exchanger system, made of two complete fin arrays shown in Figure 3.

Table 6 describes the performance of the complete system.

Parameter	Value
Inlet Temperature, guessed	60 C
Ambient Air Temperature	40 C
Outlet Temperature, to Case	48.6 C
Total Power Dissipated	276.7 W

Table 6: Heat Exchanger Performance.

The total power dissipated of 264.9 W meets our design criteria of being able to dissipate at least 250 W. The ambient air temperature is based on the design requirement that the case be able to operate in 40 C and the fan rate is given in standard units for a typical case fan, in this case an Antec TriCool 120⁹ case fan on the medium speed setting. The inlet temperature was guessed base on the output of the CPU heatsink, which is the most significant contribution of heat to the inside of the case.

3.3 CPU Heatsink

3.3.1 Investigation of Heat Pipes

The next important component to design was the heatsink that dissipates the heat from the highest single load in the case, the CPU. Several methods were investigated to do this.

Since the heat load from the CPU comes from a very small area heat pipes would be a very effective means of moving heat to a larger area where a low speed fan could move that heat into the air stream of the case. Heat pipes also minimize the thermal resistance, as the evaporator and condenser ends are nearly isothermal. The disadvantage of heat pipes is that they are difficult and thus expensive to fabricate, and have finite limits on how much heat that they can carry.

The amount of heat that a heat pipe can carry is governed by various different limits, which occur at different temperatures. When designing a heat pipe it is hard to know what limit is governing the heat consumption without calculating them all. This is shown by a typical heat pipe limit graph in Figure 4.

⁹ Antec. Product Flyers. 2006. <<http://www.antec.com/pdf/flyers/TriCool.pdf>> (16 December 2007)

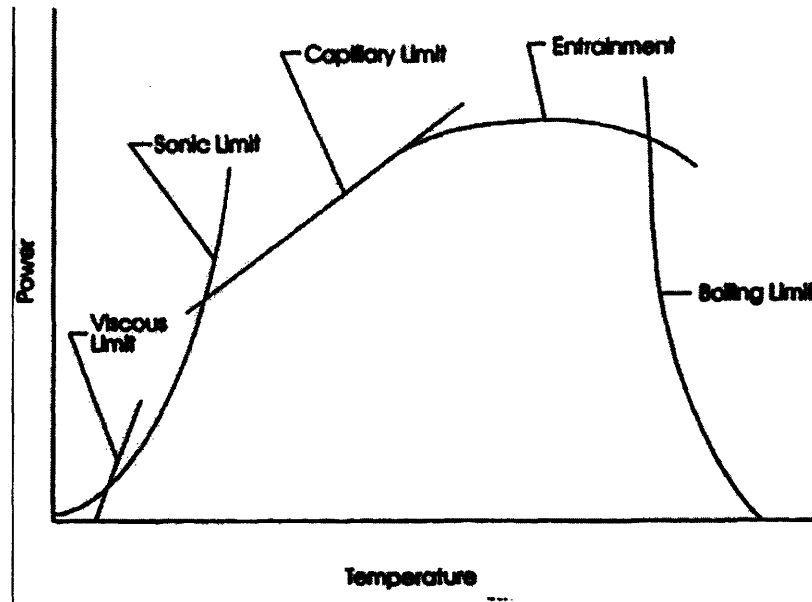


Figure 4: Heat Pipe Limit Graph¹⁰

The highest of these limits, at sub-boiling fluid temperatures, as is the case with this design, is the capillary limit. Calculating this limit will help determine whether the use of heat pipes is a viable option for removing heat from the CPU. This limit is described by Equations 12 and 13.

First, two factors related to the capillary limit are calculated. These are given in Equation 12.

$$\varepsilon = \frac{1.05\pi N d_w}{4} \quad (12a)$$

$$K = \frac{d_w^2 \varepsilon^3}{122(1 - \varepsilon)^2} \quad (12b)$$

where d_w is the diameter of wire in the mesh, N is the mesh number, the number of holes in the mesh per square meter. An additional factor, C , is a property of the pipe geometry and is given by correlation.

¹⁰ G.P. Peterson, Introduction to Heat Pipes: Modeling, Testing, and Applications 1st ed. (Wiley, New York, 1994), p.124

$$\frac{2\sigma\cos(\theta)}{r_c} = \left[\frac{C\mu_v R_v}{2\left(\frac{d_v}{2}\right)^2 A_v \rho_v h_{fg}} \right] L_{eff} q + \left[\frac{\mu_l}{KA_w h_{fg} \rho_l} \right] L_{eff} q + \rho_l g d_v \quad (13)$$

where θ is the wetting angle, σ is the surface tension of the working fluid, r_c is capillary radius or 1 over 2 times the mesh number, C and K are the factors described earlier, d_v is diameter which vapor passes through, ρ_v is the density of the vapor, μ_v is the viscosity of the vapor, h_{fg} is the heat of evaporation of the liquid, L_{eff} is the effective length, which is the adiabatic length added to half of the evaporator length, and half of the condenser length, A_v is the cross sectional area that vapor can pass through, A_w is the cross sectional area that is taken up by the layers of wire mesh, ρ_l is the density of the liquid, μ_l is the viscosity of the liquid, g is acceleration of gravity, R_v is a factor related to the wire geometry, and q is the amount of power that can be dissipated from each heat pipe.

Solving for q gives the amount of power that can be dissipated through one pipe. For this sample design a simple copper wire mesh was chosen with the following properties, shown in Table 7.

Property	Value	Unit
Mesh No.	3.937E3	1/m ²
Layers	4	
Wire Diameter	1.143E-4	m
Evaporator Length	0.015	m
Condenser Length	0.096	m
Adiabatic Length	0.020	m
Pipe Inner Diam.	0.01	m
Spacing (b/t layers)	1.143E-4	m
Wall Thickness	0.003	m

Table 7: Heat Pipe Design Properties For A Simple Test Case

More advanced geometries, while they may dissipate more heat would be very costly to fabricate. The result of this sample design gave a capillary limit of 28.22 W. This means that 5 heat pipes would be required to dissipate the power from the hottest CPU's making this an ineffective design cost-wise.

3.3.2 Pin Fin Heatsink

A cheaper, yet very effective method would be to use a pin fin heat sink. This can be made cheaply, by deep forging, and can be just as effective at removing heat from the CPU. Pin fin correlations were obtained from *Extended Surface Heat Transfer* by Kraus, Aziz, and Welty¹¹.

As there are many fins in close proximity with each other on the array. Correlations are needed to find the maximum velocity in the array. These are based on array geometry are given in Equation 14.

$$V_{\max} = \frac{S_t}{S_t - d_{pin}} V \quad (14a) \quad , \quad C = 0.35 \left(\frac{S_t}{S_l} \right)^{1/5} \quad (14b)$$

where V is the velocity of air from the fan, d_{pin} is the diameter of each pin, and S_t and S_l are factors based on the geometry, which is given in Figure 5.

¹¹ A.D. Kraus, A. Aziz, and J. Welty, Extended Surface Heat Transfer 1st ed. (Wiley, New York, 2001), p.178-180

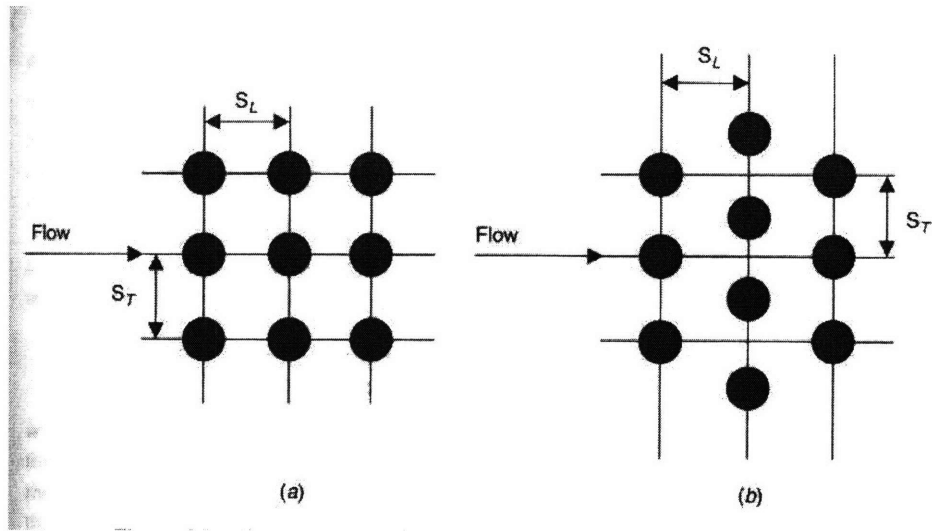


Figure 5: Standard Arrangement Of Pin Fins in an Array¹²

The convection coefficient h is determined with a Nusselt number based on the diameter of the pin fins. This is given in Equation 15.

$$Nu = C Re^m Pr^{0.36} \quad (15)$$

where C is the factor calculated earlier, m is exponent related to the placement of the pins, which is given by a table¹³, Re is a Reynolds number based on the diameter of the pins and the velocity calculated in Equation 14a, and Pr is the Prandtl number of air.

Next, the factors m and M , given by Equation 8 were calculated and plugged into Equation 9 to obtain the admittance for a single pin fin.

In order to get the heat from a small source to a larger area a square-shaped heat spreader made of highly conductive copper was used. This square can be modeled as 8 identical

¹² A.D. Kraus, A. Aziz, and J. Welty, Extended Surface Heat Transfer 1st ed. (Wiley, New York, 2001), p.179

¹³ A.D. Kraus, A. Aziz, and J. Welty, Extended Surface Heat Transfer 1st ed. (Wiley, New York, 2001), p.178

pieces connected with isothermal boundaries, across which no heat flows. Figure 6 shows these boundaries.

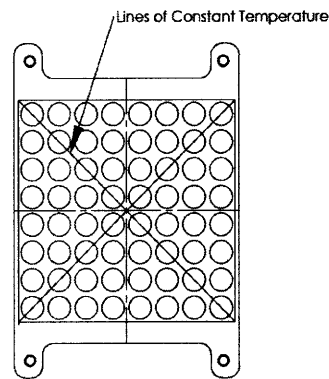


Figure 6: Lines of Constant Temperature in a Square Heat Spreader

These plates have a $1/8$ of the total number of fins on them from which to dissipate heat. If a small slice, of thickness dx , of one of the plates is taken then the number of fins on the section increases as we go down the length of the section and the total base area becomes larger. The slice of the heat spreader plus the fins on top become a composite fin that can be modeled by Equations 8 and 9. Figure 7 shows a diagram of a slice of this composite fin.

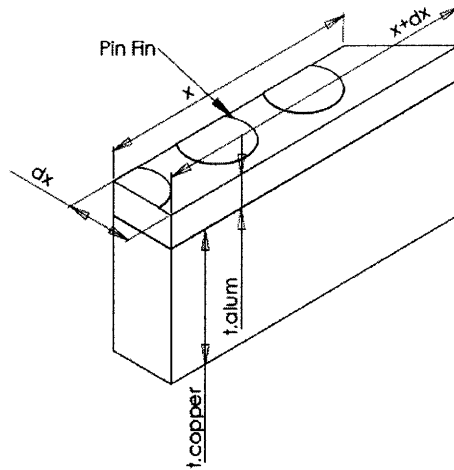


Figure 7: Slice of Composite Fin Of Thickness dx

In this case heat is assumed to come from a point in the center of the composite fin shown in Figure 6. Like a traditional fin heat enters by conduction at the side of length x and exits by conduction at side $x+dx$ and through convection to the outside. The effective convection coefficient h_{eff} , is represented by the total admittance of the pin fins on the surface, divided by the total area of the base. The pin fins themselves are not included in the composite fin, but influence the effective convection. This is described by Equation 16.

$$h_{eff} = \frac{Y_{fin} n}{A_{base}} \quad (16)$$

where Y_{fin} is the admittance of one pin fin, n is the total number of fins on the base, and A_{base} is the area of the base, which is bounded by the square in Figure 6. This value has units of W/m^2K , which are the same units as a convective heat transfer coefficient.

The wetted perimeter, P , is the length along the top surface of the composite copper/aluminum section of width dx . In order to obtain the full perimeter integration is performed along the length of the section as shown by Equation 17. Because each

triangular section between the isothermal boundaries has half the area of a square of the same dimensions a coefficient of $\frac{1}{2}$ is placed before the dx .

$$P = \int_0^{L_{side}/2} \frac{1}{2} dx = \frac{L_{side}}{4} \quad (17)$$

where L_{side} is the length of one side of the square. This is also the the average height of the triangular section bounded by the isothermal lines. The last parameter we need is the effective conductivity of the composite fin. To obtain this we take a weighted average of the total conductivity based on the length the heat has to conduct through, which would be the thickness of each metal, in this case copper on the bottom to spread the heat quickly, and aluminum on the top to attach to the pin fins. This leads to Equation 18, which describes the effective conductivity.

$$k_{eff} = \frac{t_{Cu}}{t_{tot}} k_{Cu} + \frac{t_{Al}}{t_{tot}} k_{Al} \quad (19)$$

where t_{Cu} and t_{Al} are the thickness of the copper and aluminum components respectively, k_{Cu} and k_{Al} are the conductivities of the copper and aluminum, and t_{tot} is the total thickness of copper and aluminum added together.

The cross-sectional area is simply the wetted perimeter multiplied by the total thickness of the copper and aluminum blocks pieces. This is described by Equation 19.

$$A_{cx} = Pt_{tot} \quad (19)$$

With all these values solved for they are plugged into Equation 8 to get M and m , and then into Equation 9 to obtain the total admittance of the entire heatsink.

3.3.3. Design and Optimization

The optimization of the CPU heat sink was centered on maximizing heat transfer capability and keeping weight down, as well as the requirement to use a standard 96 mm computer fan. The final design is shown in Figure 8.

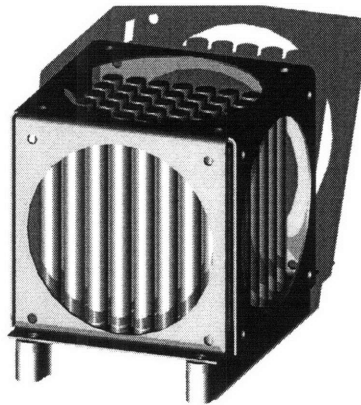


Figure 8: CPU Heatsink

Since that part is not visible to the user when the case is closed, aesthetics were not a primary concern. Table 8 shows the parameters that were used. The fan used was a Thermaltake 92mm 56CFM¹⁴ case fan.

Parameter	Value
Fan Flow Rate	56 CFM
S_t	0.012 m
S_l	0.012 m
Pin Diameter	0.01 m
Fin Length	0.096 m
Baseplate Dimensions	0.096 m x 0.096 m
Number of Fins	64
Heat Transfer Coefficient, h	242.6 W/(m ² *K)
m*L (one pin fin)	2.094
Aluminum Baseplate Thickness	0.005 m
Copper Baseplate Thickness	0.015 m
Total Admittance (for array)	10.7 W/K

Table 8: CPU Heatsink Design Parameters

¹⁴ Xoxide. Product Specification, Thermaltake 92mm case fan. 2007
<<http://www.xoxide.com/ther92casfan.html>> (16 December 2007)

The input temperature and heat flux that governed the design were based on the CPU that needed the most dissipation and the CPU that had the smallest maximum surface temperature. These values were taken from Chapter 2 and were defined as 130 W and 65 deg C respectively. The total power dissipated includes a Safety Factor of 1.25. Table 9 shows overall performance.

Parameter	Value
Input Air Temperature	48.8 C
Ambient Air Temperature	40 C
CPU Surface Temperature	64.01 C
Total Power Dissipated	162.5 W

Table 9: CPU Heatsink Performance

The heatsink was able to dissipate the required heat and still keep the CPU surface temperature below the requirements set by Intel and AMD for their chips.

3.4 Hard Drive Heatsink

The hard drive heatsink is designed much like the main case heatsink on the outside and the thermal performance is governed by the same equations. It uses natural convection and fins on the outside that maximize that convection.

The inside however is a mix of conduction on the brackets and through the front surface where more of the hard drive body contacts the heatsink. Of course this highly dominated by dissipation through the front face of the hard drive, which is a flat metal surface. The hard drive heatsink takes advantage of the way hard drives are built, and assumes all of the dissipation occurs through the front face of the disk. This is a valid assumption since the bottleneck of the heat transfer path from the hard disk to the outside is the natural convection on the outside; therefore a small increase in admittance between the hard disk and the heatsink on the inside is negligible.

It is designed to employ natural convection in the same manner as the main heat exchanger. An optimum spacing between the fins was calculated using Equation 5, and a Rayleigh Number was calculated using Equation 6. The Nusselt number for uniform heat

flux was calculated using Equation 7. The fin admittance was calculated using Equations 8 and 9.

The fin admittance takes into account the thickness of the baseplate as well, which is a simple linear conduction resistance.

3.4.1. Design and Optimization

The heatsink was designed not to protrude far from the front of the case, as it would be very visible to the user. It was also designed to accommodate a standard 3.5 inch wide hard disk. It can also accommodate smaller drives with a different mounting bracket.

Table 10 shows the design parameters and the performance of the heatsink.

Parameter	Value
Fin Thickness	0.005 m
Fin Length	0.01 m
Number of Fins	8
Separation Distance (b/t fins)	0.0059 m
Baseplate Dimensions	0.14 m x 0.092 m
Baseplate Thickness	0.001 m
m*L	0.348
Heat Dissipation Required	27 W
Total Admittance (for array)	1.926 W/K
Hard Disk Surface Temp At Startup	54.01 C
Hard Disk Surface Temp At Idle	44.15 C

Table 10: Hard Disk Heatsink Design Parameters and Performance

The heatsink was able to maintain sufficiently low temperatures even at the high heat dissipation startup condition, and lower temperatures during normal operation for 7500 and 10,000 RPM hard drives.

Chapter 4: Mechanical Design

4.1 Overall Size and Configuration and Design For Assembly

The case was made to be a tower form factor, as that would allow for an optimum orientation of the main heatsinks, which achieve best performance when the fins are oriented vertically, as the heat is free to rise naturally in ambient air. Figure 9 shows a solid model of the overall case.

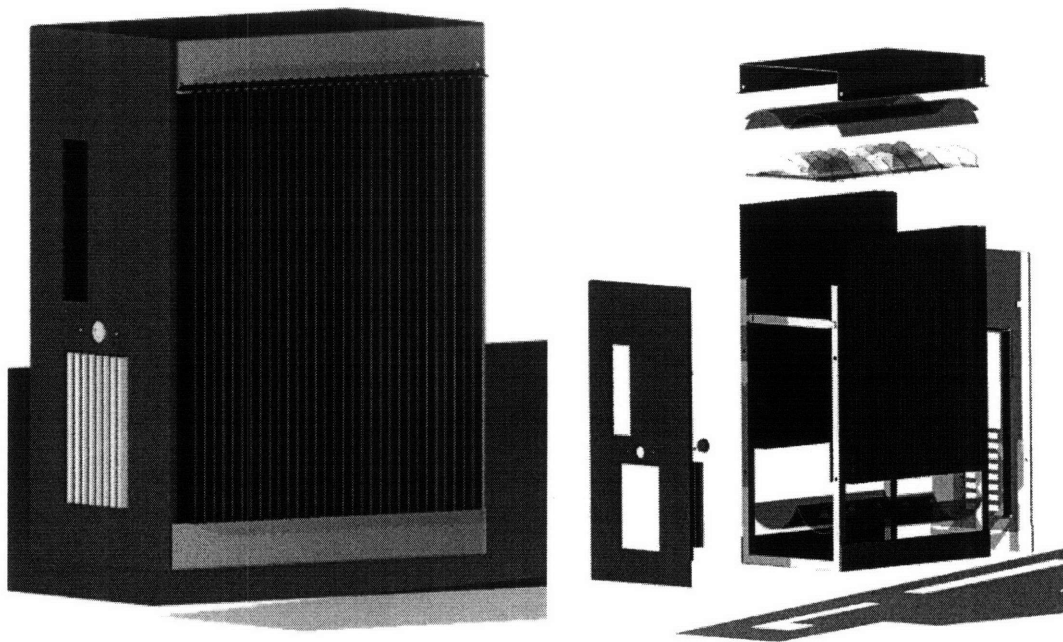


Figure 9: Complete Case (Exploded View, Right)

The overall size of the main heatsinks was selected to be 30 x 40 cm, which would allow space for a full size ATX motherboard, which would be mounted to the back of the heatsink.

The configuration was made to be easy to assemble and uses a lot of sliding fits, like for the main heat exchangers, and symmetrical parts. There are only 2 different kinds of fasteners, M4 and M7, in the case itself for ease of use.

4.2. Materials Selection

Materials were selected primarily for maximum thermal performance. Most of the main heat-dissipating components were made of 6061-T6 Aluminum. Copper was used where particularly high heat fluxes were required, such as in the CPU heat sink. Components that were not critical to heat dissipation were usually made of sheet aluminum to minimize weight and make manufacturing by bending easier. The front of the case was made of ABS plastic to minimize weight give a more pleasing look to the user. The internal components, such as the airflow directors were made of ABS plastic to utilize injection molding for the complex shapes that were used in those parts. The back of the case, which would accommodate the ports and expansion cards was made of sheet aluminum to reduce weight. The connection between this part and the rest of the case used rubber sealant to keep outside air out of the case.

4.3. Design Features

4.3.1 Weight

The case was optimized for weight leading to a weight of 37 lbs (with CPU heatsink included). This compares with Apple Computer's Power Mac G5 case, which weighed 52 lbs when fully loaded with components, water-cooled processors, and a large internal power supply. Because this case lacks these heavy components inside the case, it should be of a comparable weight when fully loaded with electronic components.

4.3.2 Airflow Directors

To enable smooth and consistent air circulation throughout the case two ABS airflow directors are located the top and bottom of the case. The airflow director at the top is a

two-part injection molded piece, shown in Figure 10. It is designed to accommodate two standard 120mm computer case fans. They can be changed to high-speed fans if the user needs additional cooling. The ribs distribute the outflow from the fans evenly across the intake for the main heat exchangers and divide the flow from each fan in half. A similarly shaped air director on the bottom takes the cold air from the exhaust of the heat exchanger and directs it upward back into the fans. This minimizes friction of the air with the walls and prevents the air from getting trapped in corners, reducing the efficiency of the fans.

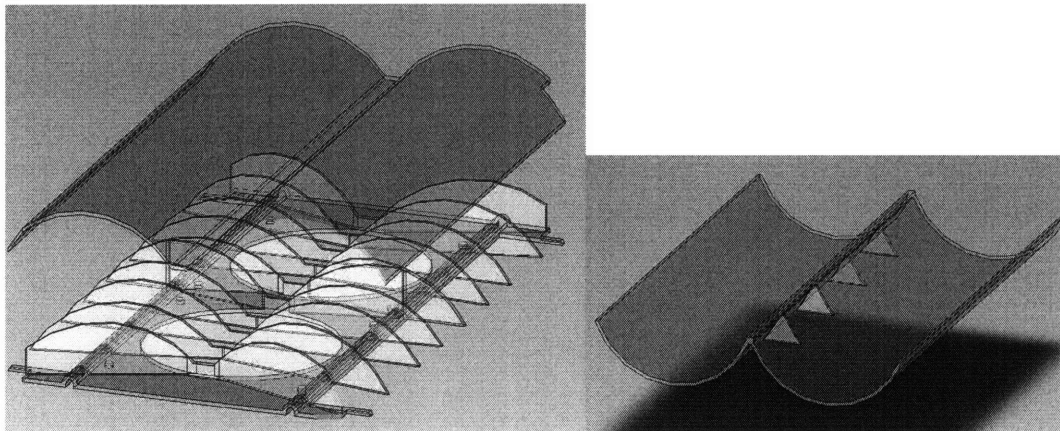


Figure 10: Airflow Directors, Top (L) and Bottom (R)

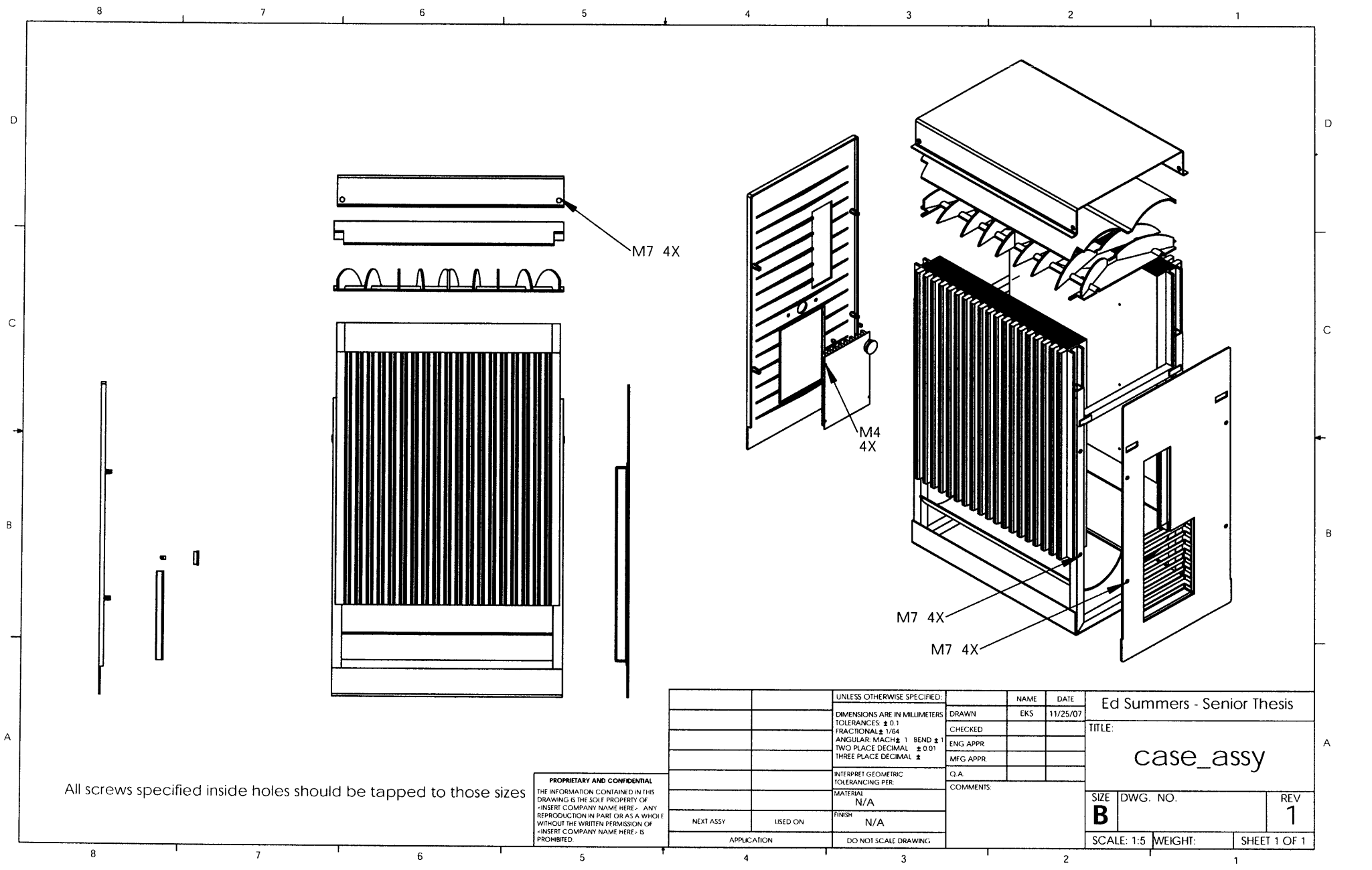
4.4. Manufacturing

The case is designed to be easily mass-produced. Plastic parts such as the airflow directors and front cover were designed to be injection molded. The CPU heatsink parts are designed to be deep forged, and the large heat exchangers to be made by metal extrusion. Other parts, such as the frame and covers are made from easy to find sheet Aluminum that can be bent into the desired shape. Spray rubber seals any openings left by bending to keep the case isolated. These manufacturing methods are only economical on a large scale; therefore making a test prototype would be rather expensive.

Chapter 5: Conclusion

The case design successfully accomplished its goals. It is able to dissipate a total of 277 W without using water-cooling or exchanging air with the outside environment. The design allows greater flexibility for users who build computers or machine integrators who need to place a PC in dirty industrial environment. Further work on the design could include experimental validation of the effectiveness of some of the components, like the large heat exchangers, or the CPU heatsink.

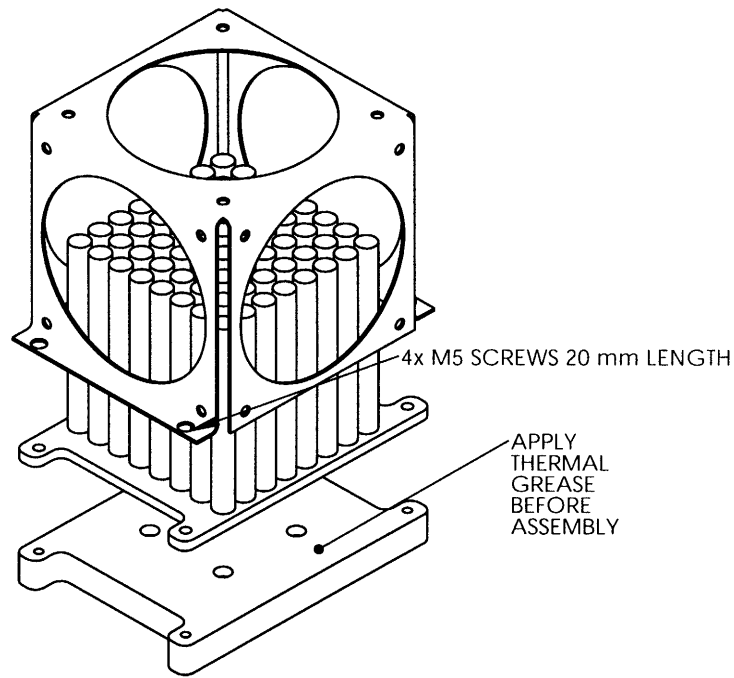
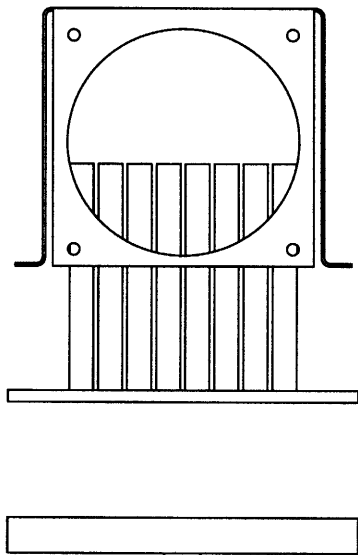
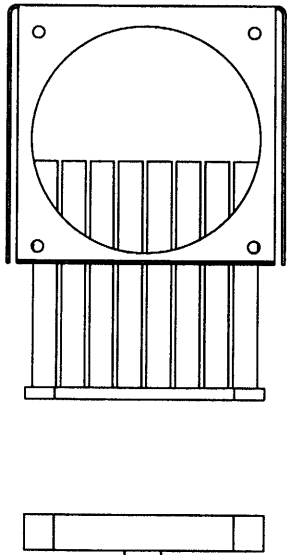
Appendix A: Engineering Drawings



All screws specified inside holes should be tapped to those sizes

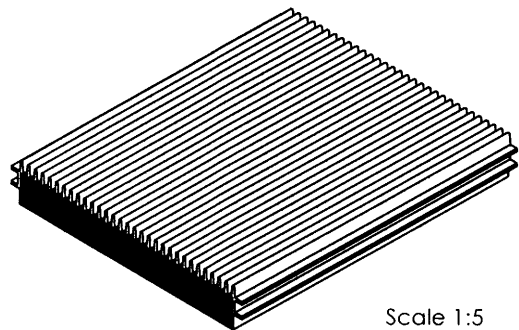
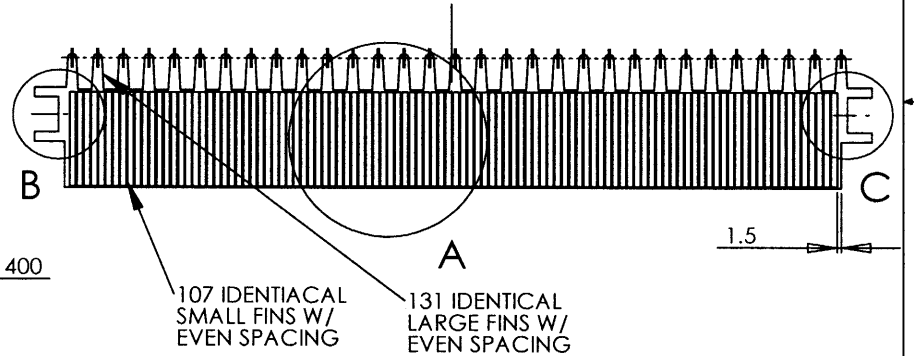
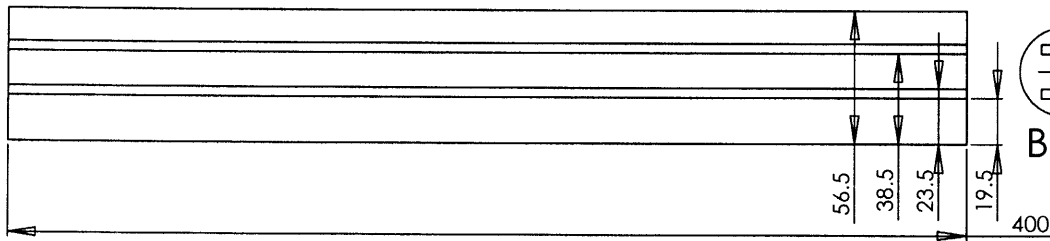
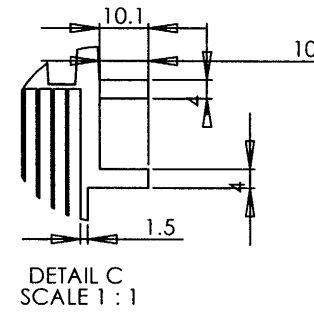
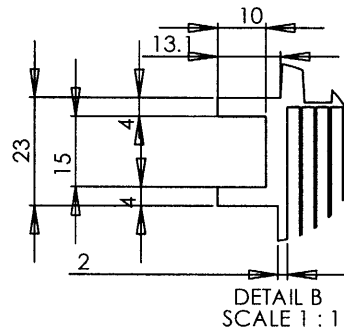
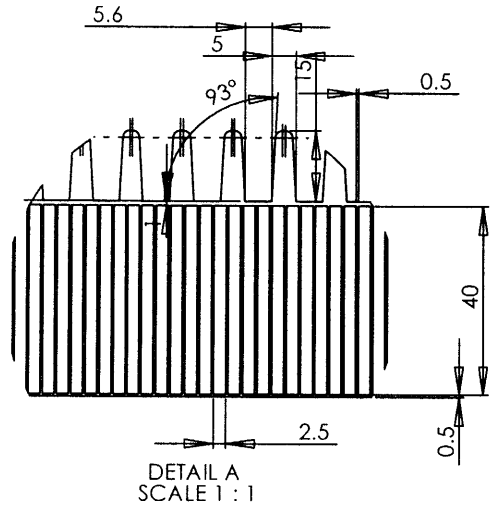
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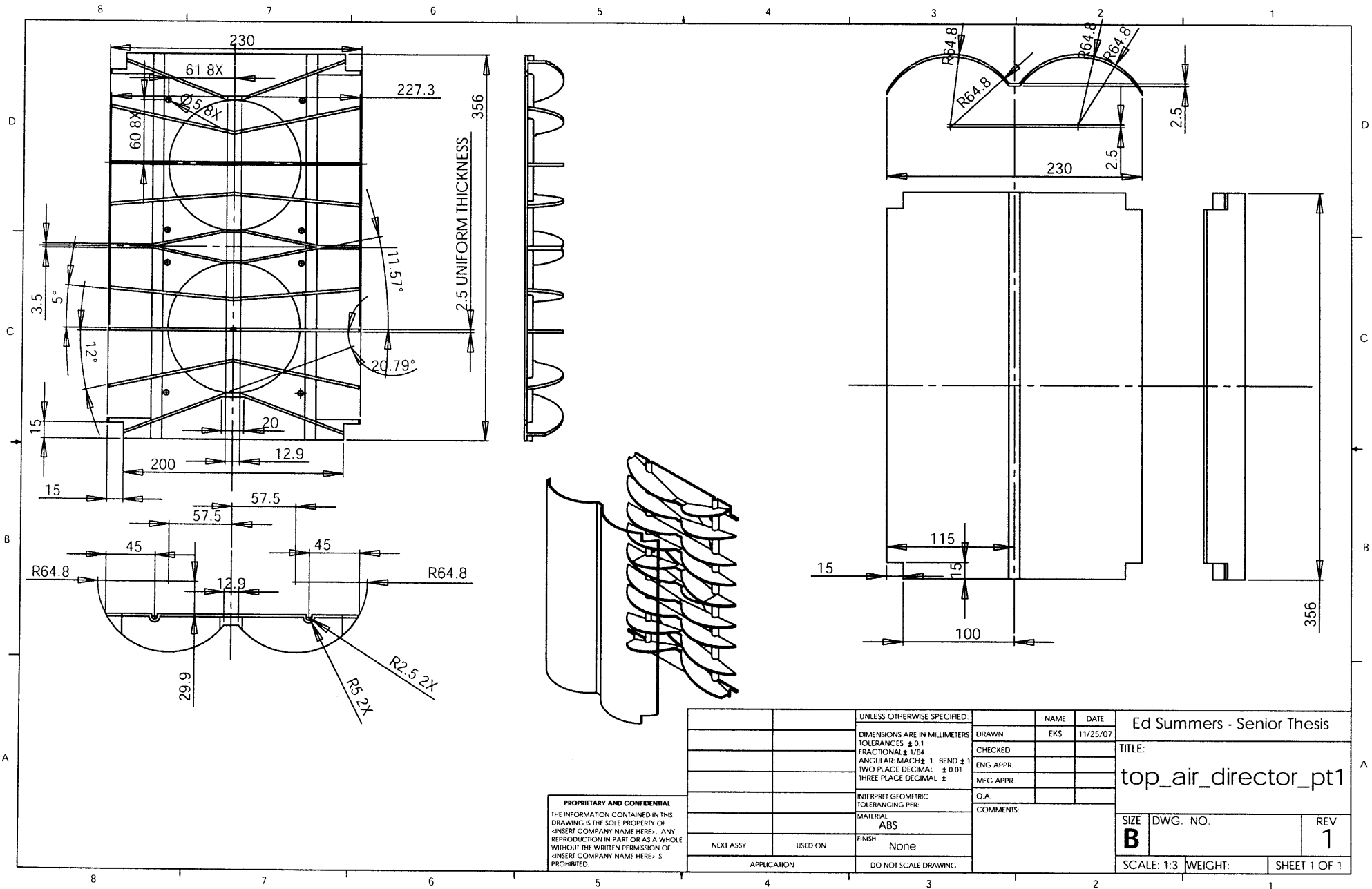
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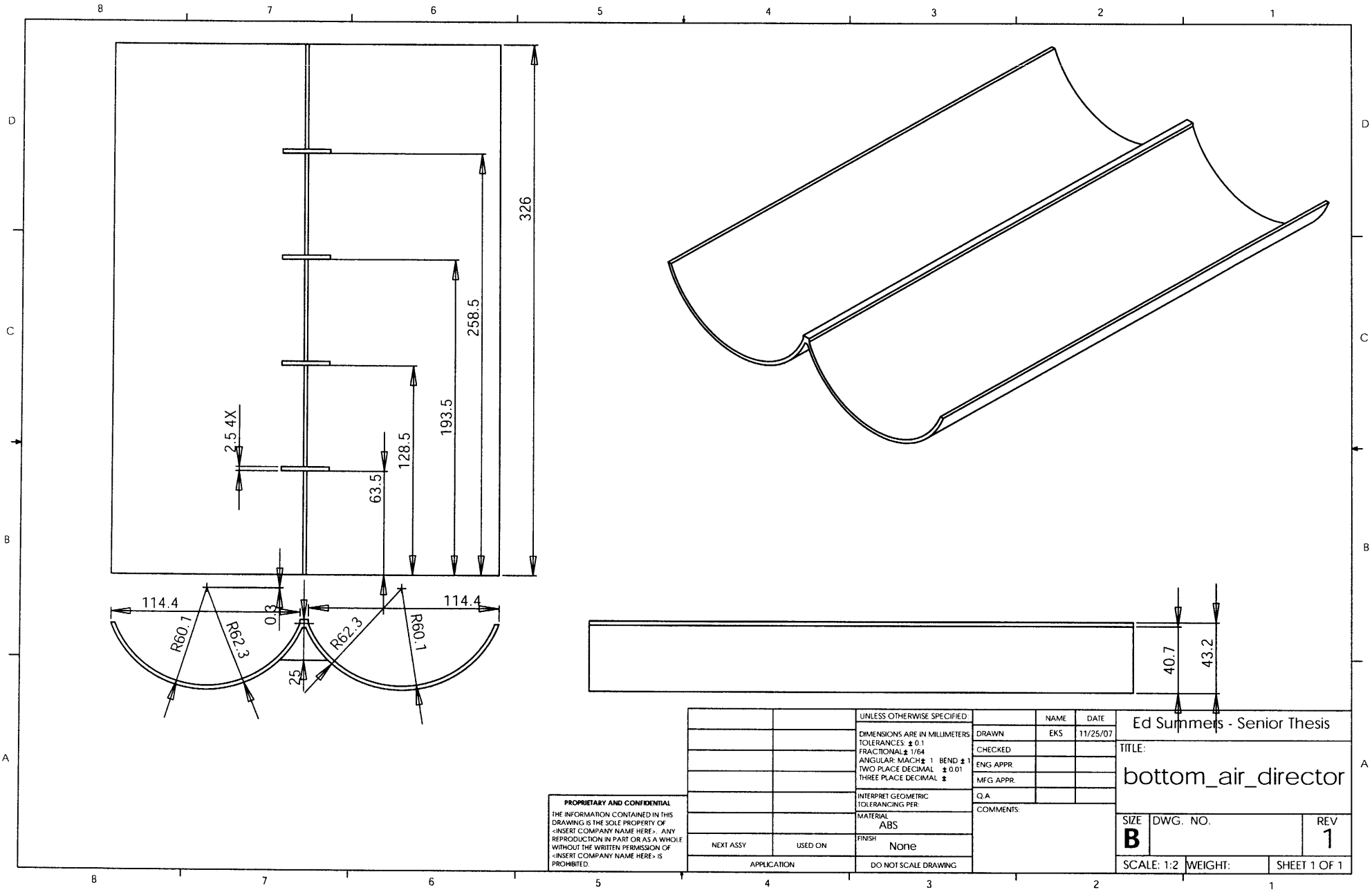
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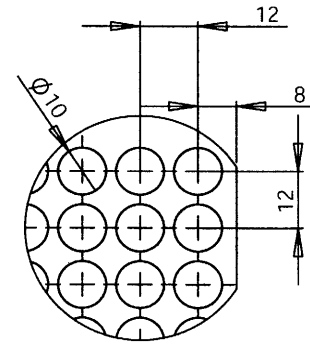
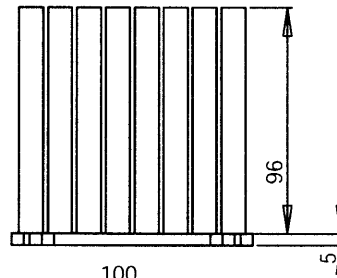
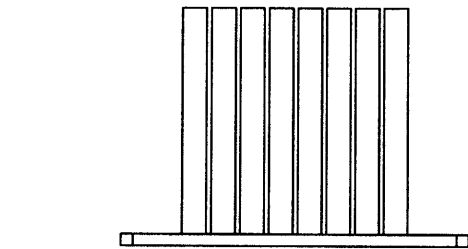
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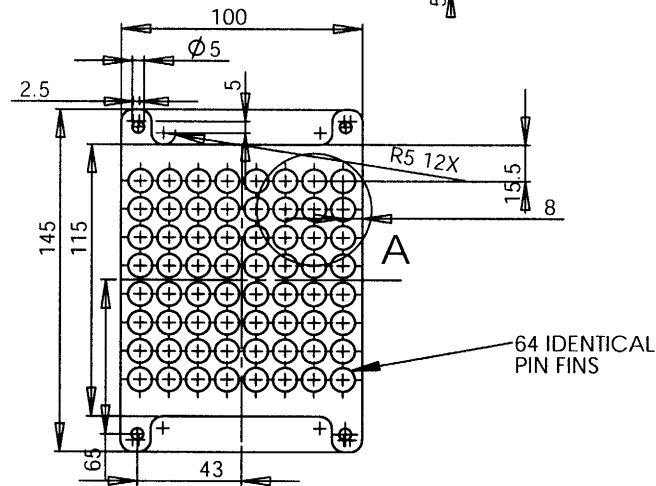
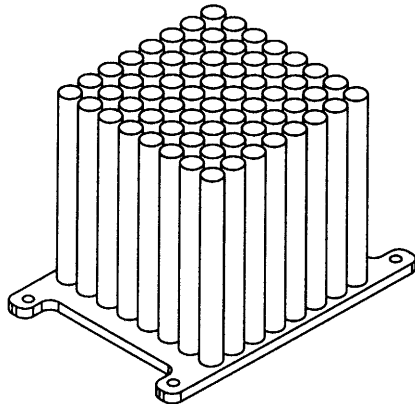
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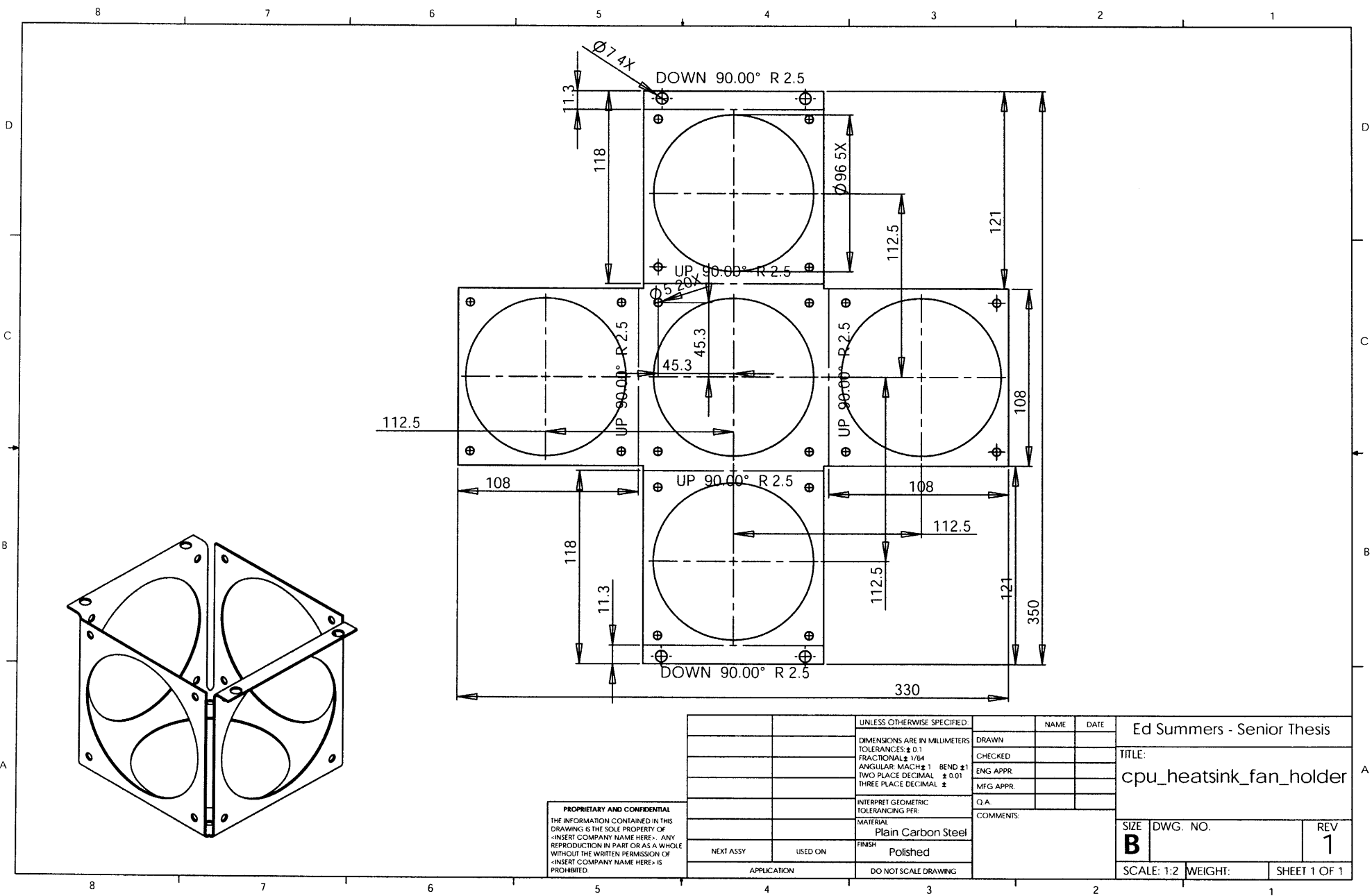


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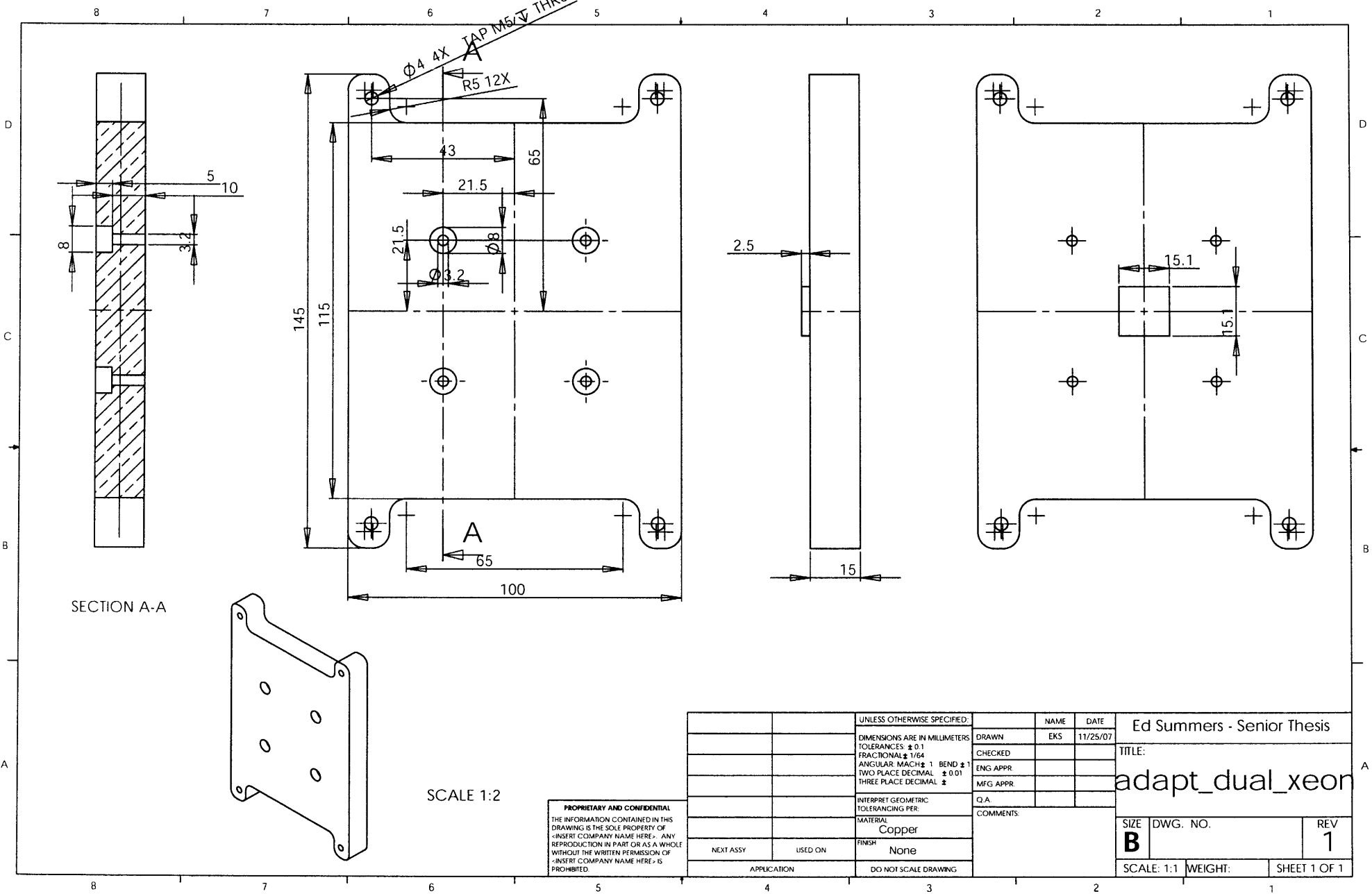
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		DIMENSIONS ARE IN MILLIMETERS		DRAWN		TITLE:	
		TOLERANCES: ± 0.1		CHECKED		cpu_heatsink_fan_holder	
		FRACTIONAL: 1/64		ENG APPR		SIZE DWG. NO.	
		ANGULAR: MACH ± 1 BEND ± 1		MFG APPR		REV	
		TWO PLACE DECIMAL ± 0.01		Q.A.		1	
		THREE PLACE DECIMAL ±		COMMENTS:		SCALE: 1:2 WEIGHT:	
		INTERPRET GEOMETRIC TOLERANCING PER:					
		MATERIAL					
		Plain Carbon Steel					
		FINISH					
		Polished					
NEXT ASSY		USED ON					
APPLICATION		DO NOT SCALE DRAWING		SHEET 1 OF 1			

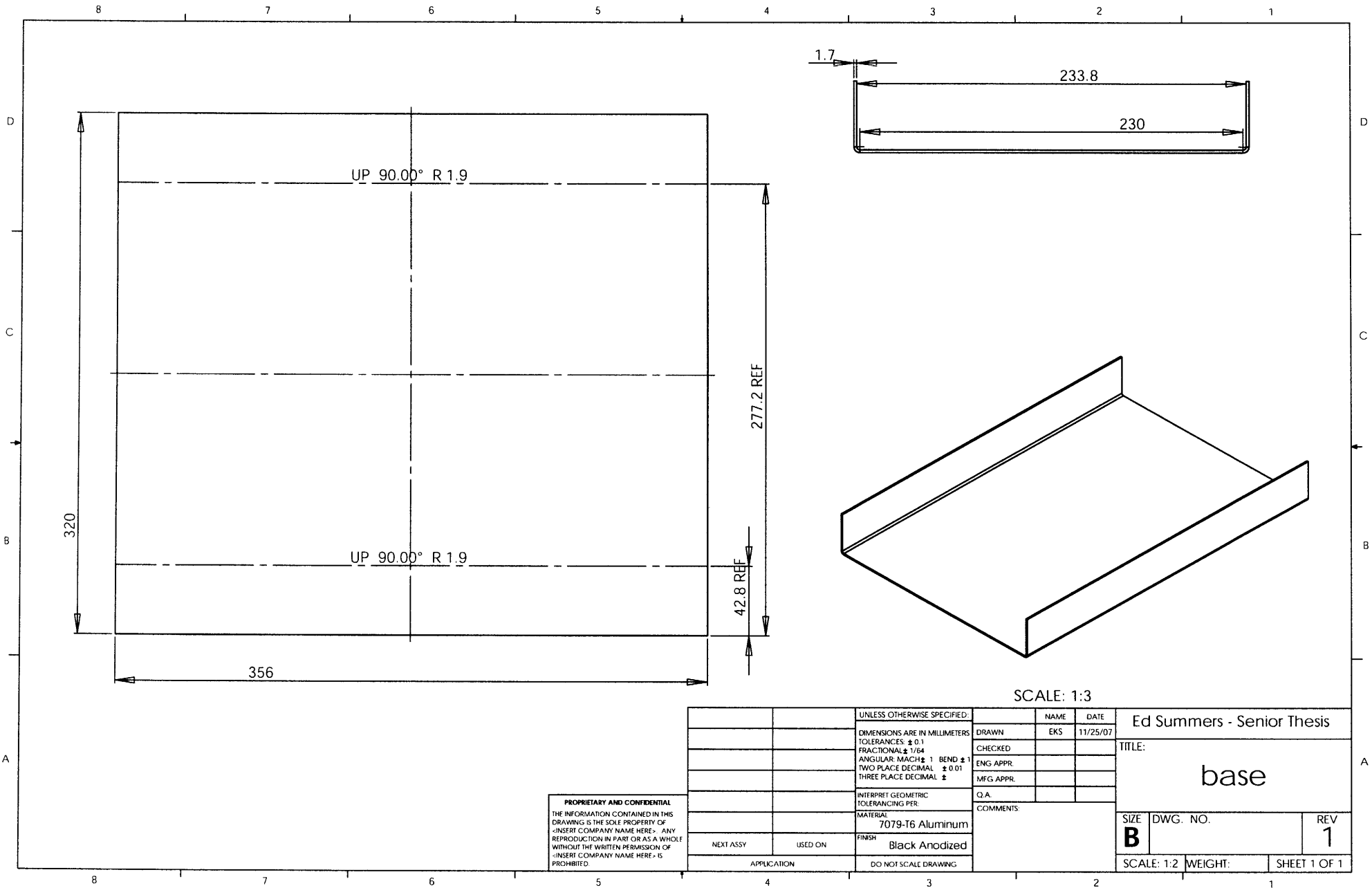


SECTION A-A

SCALE 1:2

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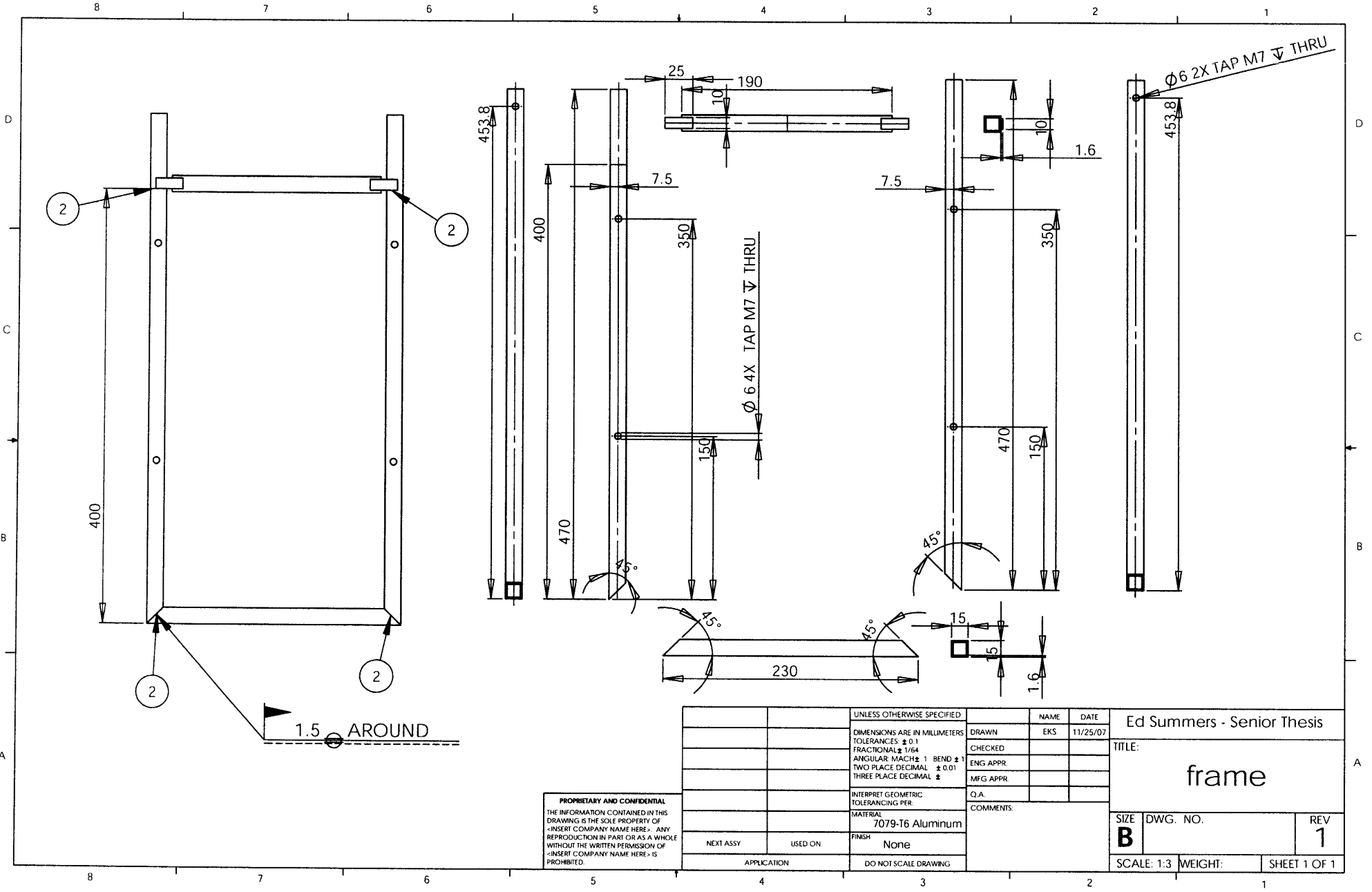
		UNLESS OTHERWISE SPECIFIED:	NAME	DATE	Ed Summers - Senior Thesis
		DIMENSIONS ARE IN MILLIMETERS	DRAWN	EKS	
		TOLERANCES: ± 0.1	CHECKED		TITLE:
		FRACTIONAL ± 1/64	ENG APPR.		adapt_dual_xeon
		ANGULAR: MACH ± 1 BEND ± 1	MFG APPR.		
		TWO PLACE DECIMAL ± 0.01	Q.A.		
		THREE PLACE DECIMAL ±	COMMENTS		
		INTERPRET GEOMETRIC TOLERANCING PER:			
		MATERIAL	Copper		
NEXT ASSY	USED ON	FINISH	None		
APPLICATION		DO NOT SCALE DRAWING			
SIZE	DWG. NO.	REV			
B		1			
SCALE: 1:1	WEIGHT:	SHEET 1 OF 1			



SCALE: 1:3

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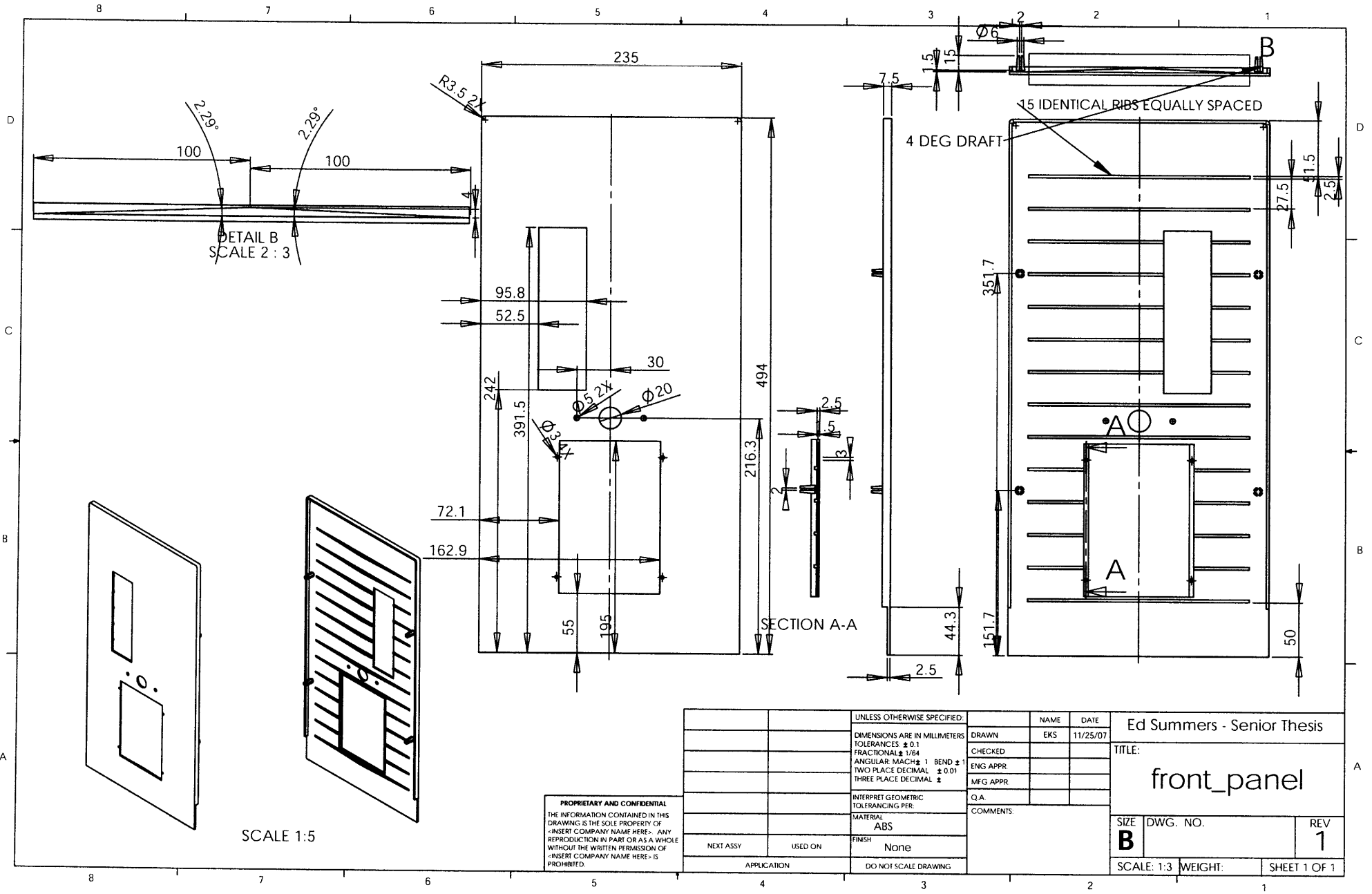
		UNLESS OTHERWISE SPECIFIED:	NAME	DATE	Ed Summers - Senior Thesis
		DIMENSIONS ARE IN MILLIMETERS	DRAWN	EKS	
		TOLERANCES: ± 0.1	CHECKED		TITLE: base
		FRACTIONAL ± 1/64	ENG APPR.		
		ANGULAR: MACH ± 1 BEND ± 1	MFG APPR.		
		TWO PLACE DECIMAL ± 0.01	Q.A.		SIZE DWG. NO. REV B 1
		THREE PLACE DECIMAL ±	COMMENTS:		
		INTERPRET GEOMETRIC TOLERANCING PER:			SCALE: 1:2 WEIGHT: SHEET 1 OF 1
		MATERIAL:			
		7079-T6 Aluminum			
		FINISH: Black Anodized			
		APPLICATION: DO NOT SCALE DRAWING			
		NEXT ASSY	USED ON		



1.5 AROUND

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		DIMENSIONS ARE IN MILLIMETERS		DRAWN	EKS	11/25/07	TITLE: frame
		TOLERANCES ± 0.1		CHECKED			
		FRACTIONAL ± 1/64		ENG APPR			
		ANGULAR: MACH ± 1 BEND ± 1		MFG APPR			
		TWO PLACE DECIMAL ± 0.01		Q.A.			SIZE DWG. NO. REV
		THREE PLACE DECIMAL ±		COMMENTS			B 1
		INTERPRET GEOMETRIC TOLERANCING PER				SCALE: 1:3 WEIGHT: SHEET 1 OF 1	
		MATERIAL					
		7079-T6 Aluminum					
		FINISH					
		None					
NEXT ASSY		USED ON					
APPLICATION		DO NOT SCALE DRAWING					

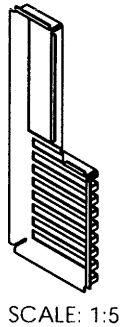
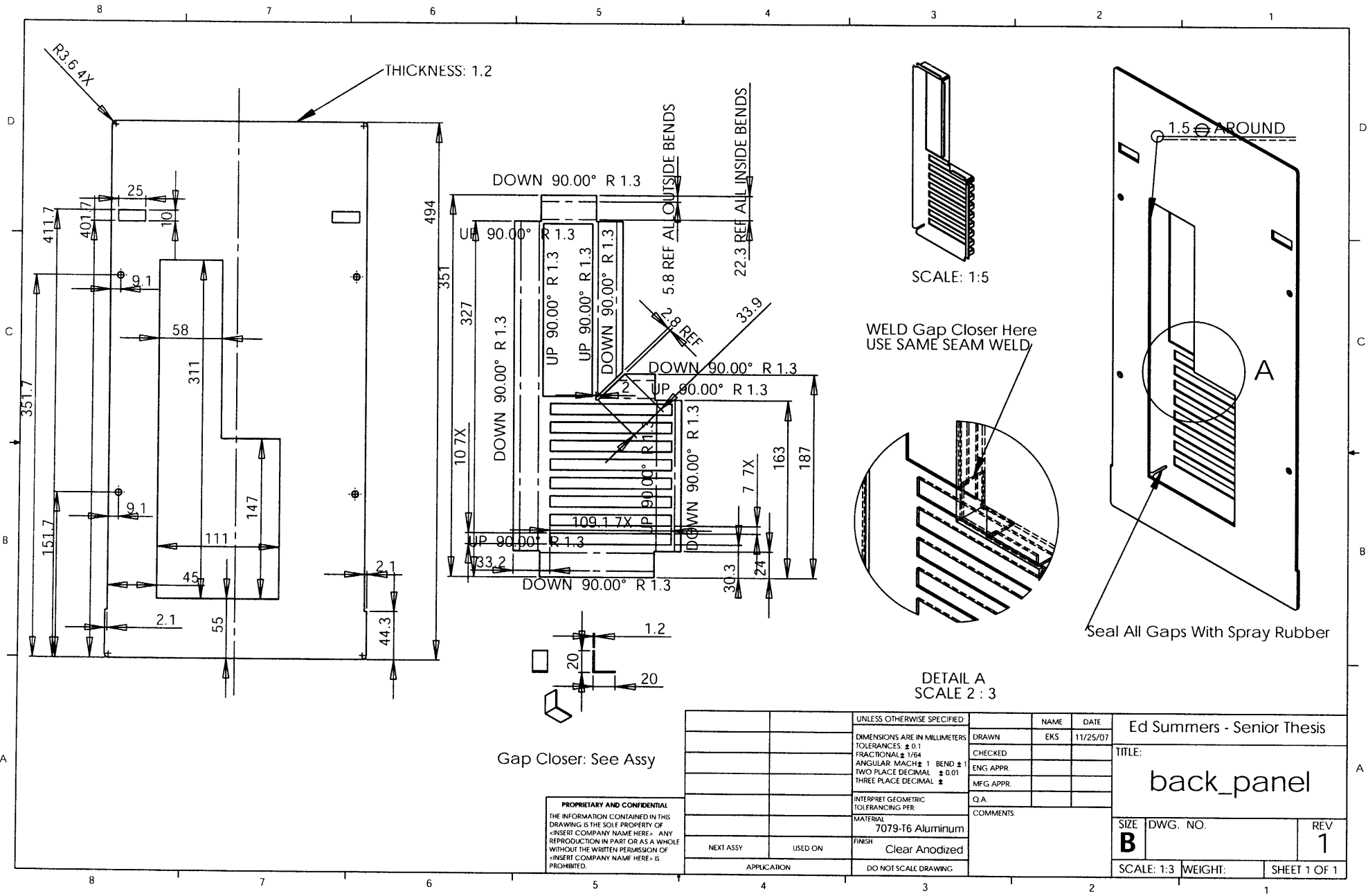


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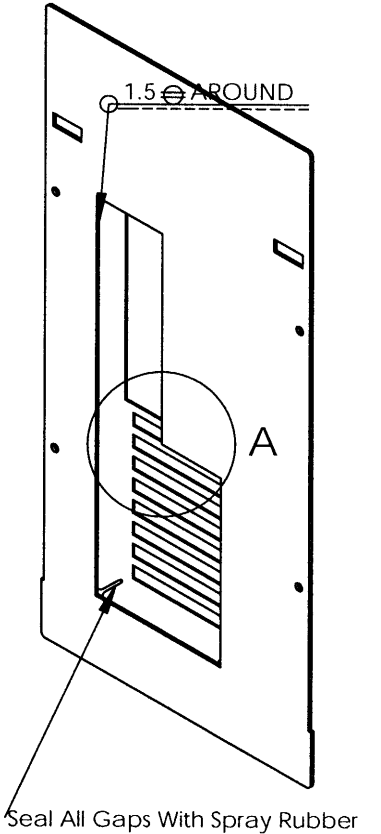
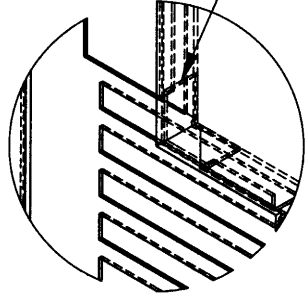
		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	Ed Summers - Senior Thesis
		DIMENSIONS ARE IN MILLIMETERS TOLERANCES ± 0.1 FRACTIONAL ± 1/64 ANGULAR: MACH ± 1 BEND ± 1 TWO PLACE DECIMAL ± 0.01 THREE PLACE DECIMAL ±		DRAWN	11/25/07	
		INTERPRET GEOMETRIC TOLERANCING PER:		CHECKED		SIZE B
		MATERIAL ABS		ENG APPR		
		FINISH None		MFG APPR		REV 1
NEXT ASSY	USED ON	APPLICATION DO NOT SCALE DRAWING		Q.A		SCALE: 1:3 WEIGHT:
				COMMENTS		SHEET 1 OF 1

SCALE 1:5

DETAIL B
 SCALE 2:3



WELD Gap Closer Here
USE SAME SEAM WELD

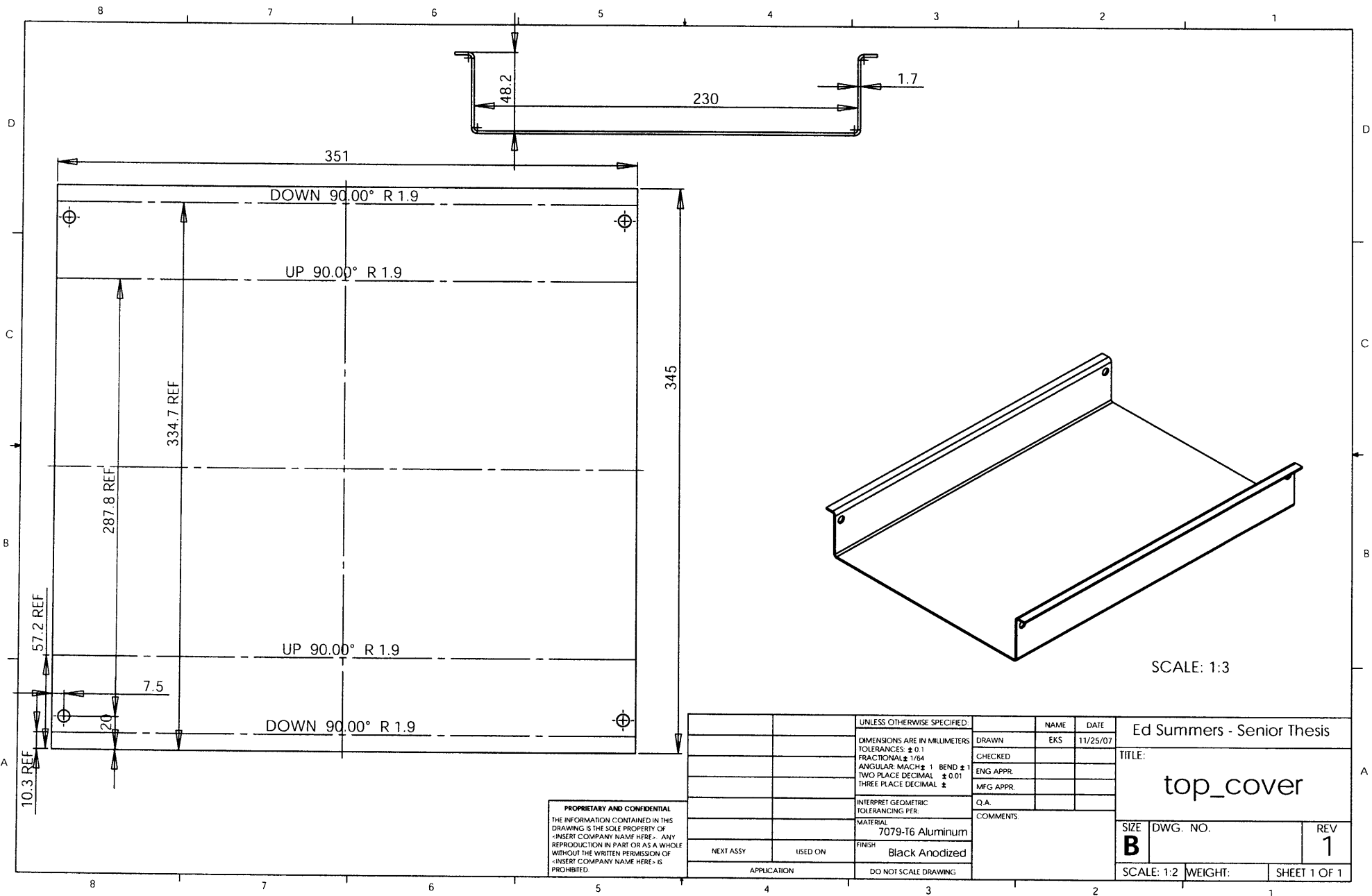


DETAIL A
SCALE 2 : 3

Gap Closer: See Assy

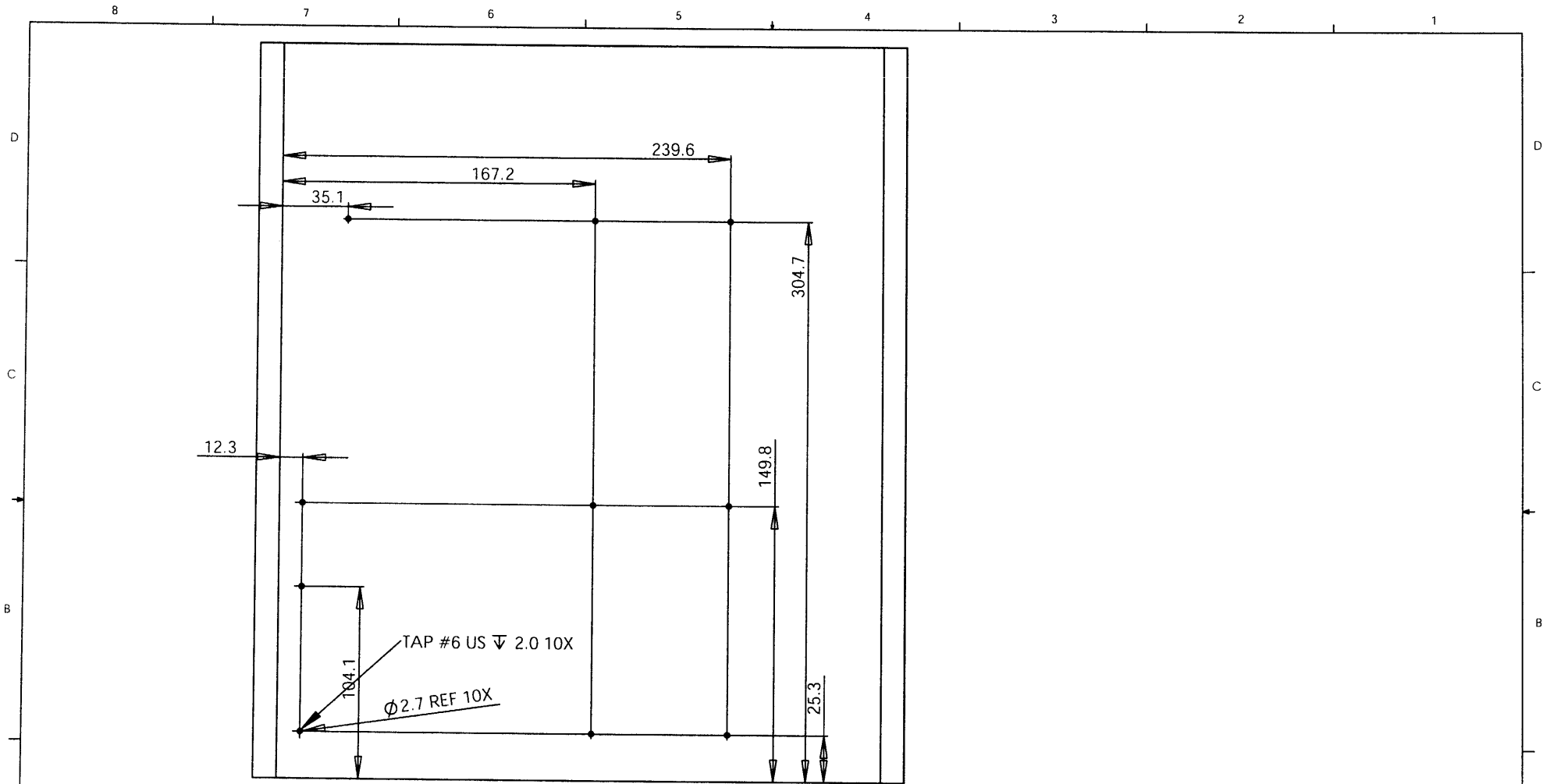
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		INTERPRET GEOMETRIC TOLERANCING PER:	CHECKED		TITLE: back_panel
		MATERIAL	ENG APPR.		
		7079-T6 Aluminum	MFG APPR.		SIZE DWG. NO.
		FINISH	Q.A.		B
		Clear Anodized	COMMENTS		REV
NEXT ASSY	USED ON	APPLICATION			1
		DO NOT SCALE DRAWING			SCALE: 1:3 WEIGHT:
					SHEET 1 OF 1



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		TOLERANCES ± 0.1	CHECKED		TITLE:
		FRACTIONAL ± 1/64	ENG APPR.		top_cover
		ANGULAR: MACH ± 1 BEND ± 1	MFG APPR.		
		TWO PLACE DECIMAL ± 0.01	Q.A.		
		THREE PLACE DECIMAL ±	COMMENTS		
		INTERPRET GEOMETRIC TOLERANCING PER:			
		MATERIAL			SIZE
		7079-T6 Aluminum			DWG. NO.
		FINISH			REV
		Black Anodized			1
NEXT ASSY	USED ON	APPLICATION			SCALE: 1:2
		DO NOT SCALE DRAWING			WEIGHT:
					SHEET 1 OF 1



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TOLERANCES: ± 0.1		CHECKED	
FRACTIONAL ± 1/64		ENG APPR	
ANGULAR: MACH ± 1 BEND ± 1		MFG APPR	
TWO PLACE DECIMAL ± 0.01		Q.A	
THREE PLACE DECIMAL ±		COMMENTS	
INTERPRET GEOMETRIC TOLERANCING PER			
MATERIAL			
6061-T6 Aluminum			
FINISH			
Black Anodized			
NEXT ASSY	USED ON		
APPLICATION			
DO NOT SCALE DRAWING			

Ed Summers - Senior Thesis		
TITLE: ATX Hole Pattern		
SIZE B	DWG. NO.	REV 1
SCALE: 1:2	WEIGHT:	SHEET 1 OF 1